DEVELOPMENT AND ANALYSIS OF

MODEL BASED ECONOMIZER STRATEGIES

By

AMIT BHANSALI

Bachelor of Science in Mechanical Engineering

Institute of Engineering and Technology, Devi Ahilya

University

Indore, Madhya Pradesh, India

2007

Submitted to the Faculty of the
Graduate College of the
Oklahoma State University
in partial fulfillment of
the requirements for
the Degree of
MASTER OF SCIENCE
December, 2009

DEVELOPMENT AND ANALYSIS OF

MODEL BASED ECONOMIZER STRATEGIES

Thesis Approved:

Dr. Daniel E. Fisher

Thesis Adviser

Dr. Jeffrey Spitler

Dr. Lorenzo Cremaschi

Dr. A. Gordon Emslie

Dean of the Graduate College

ACKNOWLEDGMENTS

There are many people whom I would like to extend my regards for helping me unconditionally and selflessly throughout my time here. It is not possible for me to express my emotions in words but I must try to express my gratitude in the best possible way I can.

First of all, I am highly thankful to God who guided me throughout my life and made it possible for me to realize my dreams. I feel that I have come in more close relationship with God during my period here.

My mother has been my source of inspiration all my life. She is the reason for what I am today and what I will be tomorrow. She cemented high values and virtues in me from my childhood and gave me courage to face any hardships. Her life remains a pillar of strength for me even after her loss nine years ago. My father has been more a friend of mine. He has always kept me focused on my goals and my dreams. He has continuously provided me with mental and emotional comfort.

It wasn’t easy to come to a different country and adapt to people and environment here. I still do remember the home sickness I felt for first couple of months. It was my family who continuously made me believe that everything would be all right and I should keep myself motivated. My sister always pushed me to do things with more intent.

She has kept me in her prayers and has kept a check on my progress.

Next to my family, the person who has made a large impact on my life is my adviser Dr. Fisher. He provided me with an opportunity to work with him and gave me a way to go ahead. He has showed his faith in me and always encouraged me to work hard. I still do remember that sometimes he has pushed me to achieve and do something which I thought I won't be able to do. He is filled with jokes and makes meetings really friendlier and eases the tension.

I would like to extend my gratitude to Johnson Controls, Inc. for giving me an opportunity to work with them during my project. My special thanks to John House and John Seem who continued to support me throughout the project work. John House has been a tremendous source of inspiration and a role model to me. His dedication to his work has been amazing to me. His input at all times has been crucial to completion of my thesis.

Dr. Spitler introduced me to the kind of hard work needed to do Masters. He is one of the best teachers I have ever had. Though he was tough but I learnt a lot from his work ethics. He made an invaluable contribution in developing my thinking process and problem approaching skills.

Dr. Cremaschi pushed me to do better that I expected of myself. I really liked his approach towards student and dedication to make them understand the subject easily. I remember that he made me work twice on my project report during my class with him. I was satisfied with the report the first time but he made me do much better than I already did.

Two of my colleagues have been more than just friends to me during this time. I feel obliged to Edwin and Sankar for being supportive, helpful and always there whenever I felt stalled. They kept my engines running with inputs and suggestions to their best possible knowledge.

I made a lot of good friends who helped me adjust and feel comfortable in an alien environment. It would be unfair to them if I don't mention their name here. Ford, Becki and Janina have served me as a true friend and have provided me with emotional and spiritual support. I am really close to their families and they have treated me as their own son. I am fortunate enough to carry these lifelong relationships from here.

I would like to express my regards to all my other friends and people here that have directly or indirectly helped me in completion of my project.

TABLE OF CONTENTS

LIST OF TABLES

Table Page

LIST OF FIGURES

Figure Page

Nomenclature

A	Area of coil
b	Bypass air fraction
C	Capacitance
C_p	Specific heat at constant pressure
ε	Effectiveness of cooling coil
EES	Engineering equation solver
f	Fraction of outdoor air
h	Enthalpy
m	Mass flow rate
NTU	Number of transfer units
Q	Load
RH	Relative humidity
small	a very small number of order (10^{-5})
tau	Golden section number (0.618)
T	Temperature
U	Overall Heat Transfer Coefficient
ω	Humidity ratio
Subscripts:	
a	air
a, in	Air side inlet
a, out	Air side outlet
ceat	Cooling coil entering air temperature
clat	Cooling coil leaving air temperature
cla	Total air leaving the coil

coil	Cooling coil
dp,oa	Dew point of outside air
dp,ma	Dew point of mixed air
fi	Fan inlet
fo	Fan outlet
FDB	Fixed dry bulb
FDP	Fixed dew point of air
FE	Fixed enthalpy of air
la	Coil contact leaving air
ma	Mixed air
max	Maximum
min	Minimum
mst	Mixed set temperature
oa	Outside air
opt	Optimum
ra	Return air
sa	Supply air
set	Set point
ta	Transition air
w	Water
wi	Water inlet
w,in	Water side inlet
wo	Water outlet
w,out	Water side outlet

CHAPTER 1

1. Introduction

Different strategies are available to control the damper movement to regulate the flow of outside air through commercial air handlers. These damper control strategies are called economizer strategies. The objective of an economizer strategy is to select a damper position that minimizes the load on the cooling coil. Some standard control strategies have been suggested by ASHRAE (90.1-2007) which are available based on different control parameters. ASHRAE also defines the climatic conditions where an economizer strategy can be used and where it can’t be used.

This chapter contains the brief description of economizer strategies that are being used traditionally and introduces the model based strategies that are the subject of this investigation.

1.1 Conventional Economizer Strategies

A schematic of a simple building management system is shown in Figure 1.1. The building management system obtains the temperature and relative humidity readings from the sensors located at outside air, return air and supply air ducts. Depending on the sensor readings and economizer strategy used, the dampers at outside air and return air inlets to the mixing chamber are controlled.

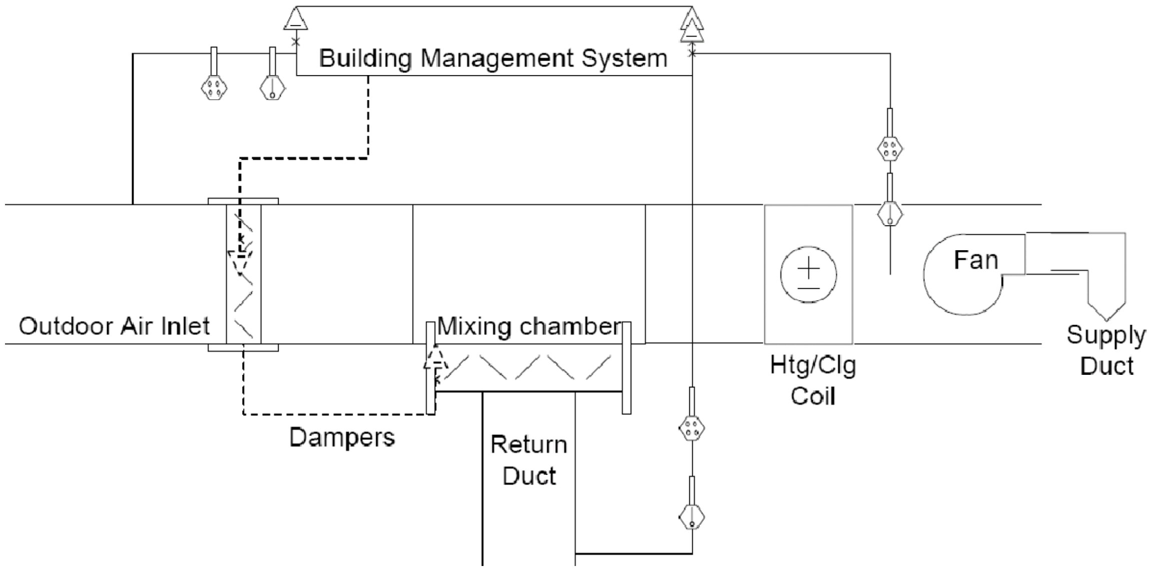


Figure 1.1: Schematic of use of economizer strategy to determine damper position

The building management system will likely have one of the following economizer strategy programmed into its logic.

1. *No economizer strategy*

   Outside air damper is maintained at closed (or minimum) position. The minimum position dictated by ventilation requirements.

2. *Differential dry bulb temperature control strategy*

   When the outside dry bulb temperature is lower than the supply air temperature, no cooling is required and outside air flow rate is increased to obtain the desired supply temperature. When outside air dry bulb temperature is more than the supply air temperature but less than the temperature of the return air then the position of the damper is varied so as to close the recirculated air duct completely and bring in full outside air. The disadvantage of controlling with this strategy is when the outside air enthalpy is more than the return air enthalpy the cooling load on the coil increases for 100% outdoor air. However, when the outdoor dry bulb temperature is more than the return air temperature then mixing is performed taking minimum amount of outside air.

3. *Differential enthalpy based control strategy*

   The difference in outdoor and return air enthalpy controls the outside air flow rate. To estimate the enthalpy, wet bulb temperature or relative humidity in addition to the dry bulb temperature is measured. The flow rate of the outside air is set to minimum when the enthalpy of the outside air is more than the enthalpy of the return air.

4. *Fixed enthalpy control strategy*

   The outside air flow rate is set to minimum if the estimated enthalpy of outside air based on temperature and humidity measurements is more than maximum value of outdoor enthalpy permissible.

5. *Fixed dry bulb strategy*

   The outside air flow rate is set to minimum if the measured dry bulb temperature of outside air is more than maximum value of outdoor dry bulb temperature permissible.

6. *Dew point temperature and dry bulb temperature control strategy*

   In addition to the dry bulb temperature, dew point temperature of the incoming outside air is compared against the allowed maximum dew point temperature. If the measured dew point temperature is more than the fixed dew point temperature than the outside air mass flow rate is set to minimum.

7. *Electronic enthalpy curve*

   This economizer strategy is governed by a curve on the psychrometric chart that passed through 75^{o}F and 40%. The curve as shown in Figure 1.3 is parallel to dry bulb temperature line for lower humidity and enthalpy line for higher humidity. If the outdoor air parameters viz. outdoor air temperature and outdoor air relative humidity is more than that on the curve, the outside air mass flow rate is set to minimum.

Economizer strategies with their governing equations are represented in Table 1.1.

Table 1.1: Economizer strategies

Control Option	Equation	Description(Economizer Off when)
Fixed Dry Bulb	$T_{oa}>T_{FDB}$	Outdoor air temperature exceeds some Fixed dry bulb temperature
Differential Dry Bulb	$T_{oa}>T_{ra}$	Outdoor air temperature exceeds return air temperature
Differential Enthalpy	$h_{oa}>h_{ra}$	Outdoor air Enthalpy exceeds return air enthalpy
Fixed Enthalpy	$h_{oa}>h_{FE}$	Outdoor air Enthalpy exceeds some Fixed Enthalpy
Electronic Enthalpy	$(T_{oa}/RH_{oa})>A$	Outdoor air temperature/Relative humidity exceeds A*
Dew Point and Dry Bulb Temperatures	$T_{dpoa}>T_{FDP}$, $T_{oa}>T_{FDB}$	Outdoor air dew point exceeds some fixed dew point or Outdoor air temperature exceeds some fixed dry bulb temparture.

*Setpoint "A" corresponds to a curve on the Psychrometric chart that goes through a point at approximately 75°F and 40% RH and is nearly parallel to Dry bulb lines at low humidity and nearly parallel to enthalpy lines at high humidity levels

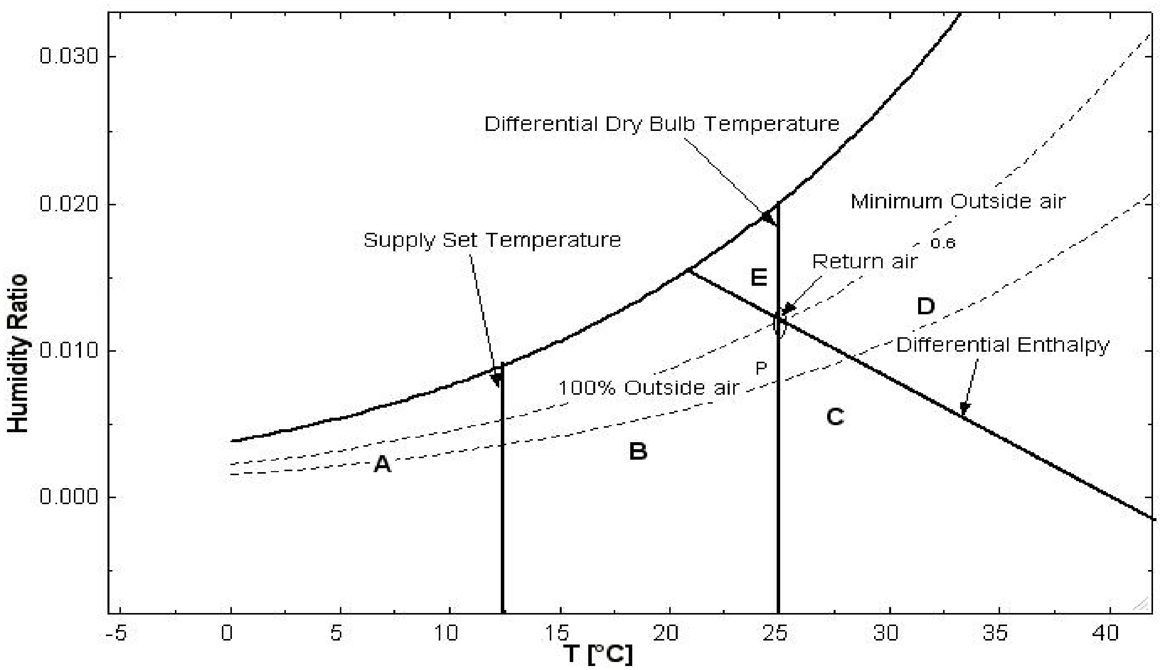


Figure 1.2: Psychrometric chart showing differential dry bulb and differential enthalpy strategies

For better understanding all the economizer strategies are shown with the help of psychrometric charts in Figure 1.2 and Figure 1.3. All the points inside the psychrometric charts represent possible outdoor air conditions. Point P represents the return air conditions. Decisions based on economizer strategies are shown in regions B, C, D, E and F. Figure 1.2 shows the two conventional control strategies that uses return air conditions as well to make a control decision for damper position. Outdoor air conditions lying in region A have temperature less than the supply set point temperature, thus, a proper fraction of outdoor air is mixed to the return air to bring the air temperature to supply set point. The damper control decision based in this region is same irrespective of any economizer strategy. However, to avoid freezing of the coil, minimum outdoor air is mixed with return air when the temperature of outdoor air is less than a specified/ recommended low temperature limit. The vertical line passing through point P in Figure 1.2 represents the dry bulb temperature of return air. The angled line represents the return air enthalpy line. Differential dry bulb strategy would allow 100% outside air in region B and E. Dampers would be adjusted to minimum outdoor air position in regions C and D for temperature based economizer. Differential enthalpy strategy would allow 100% outdoor air in regions B and C. Region D and E would represent minimum outdoor air for enthalpy based economizer. It can easily be inferred that in regions C and E, both the strategies would make different decisions.

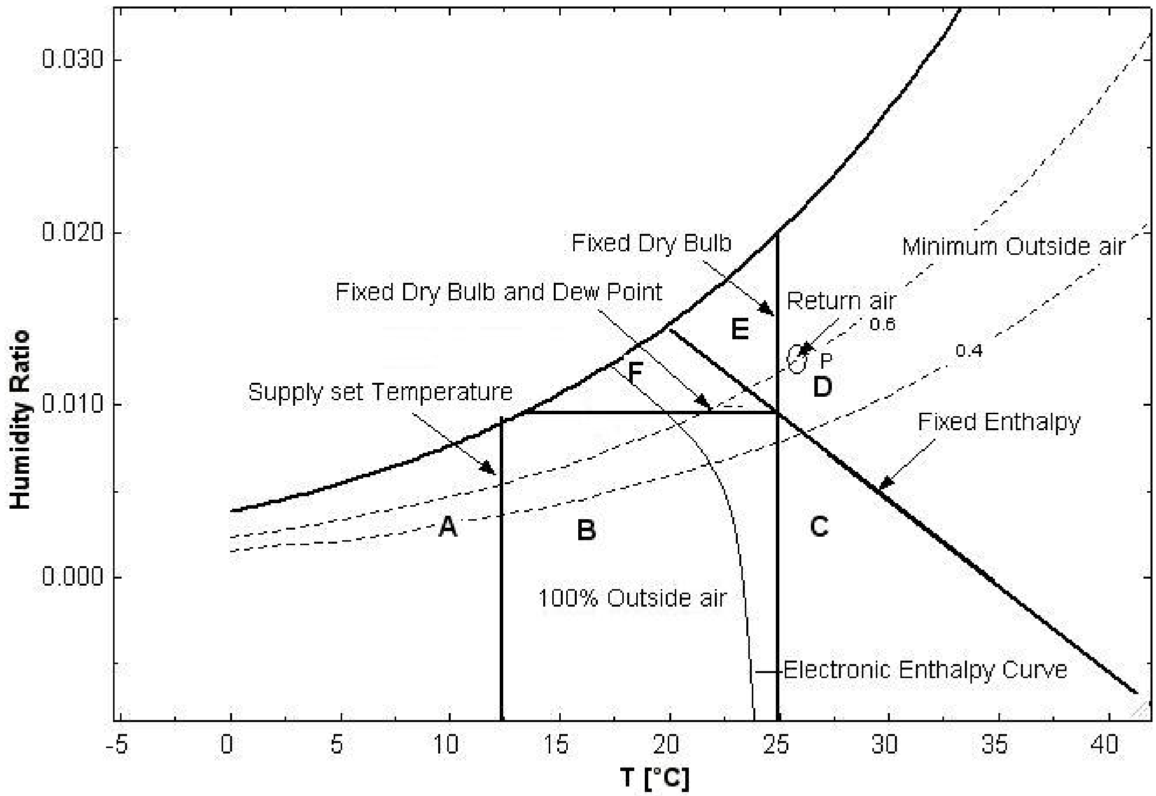
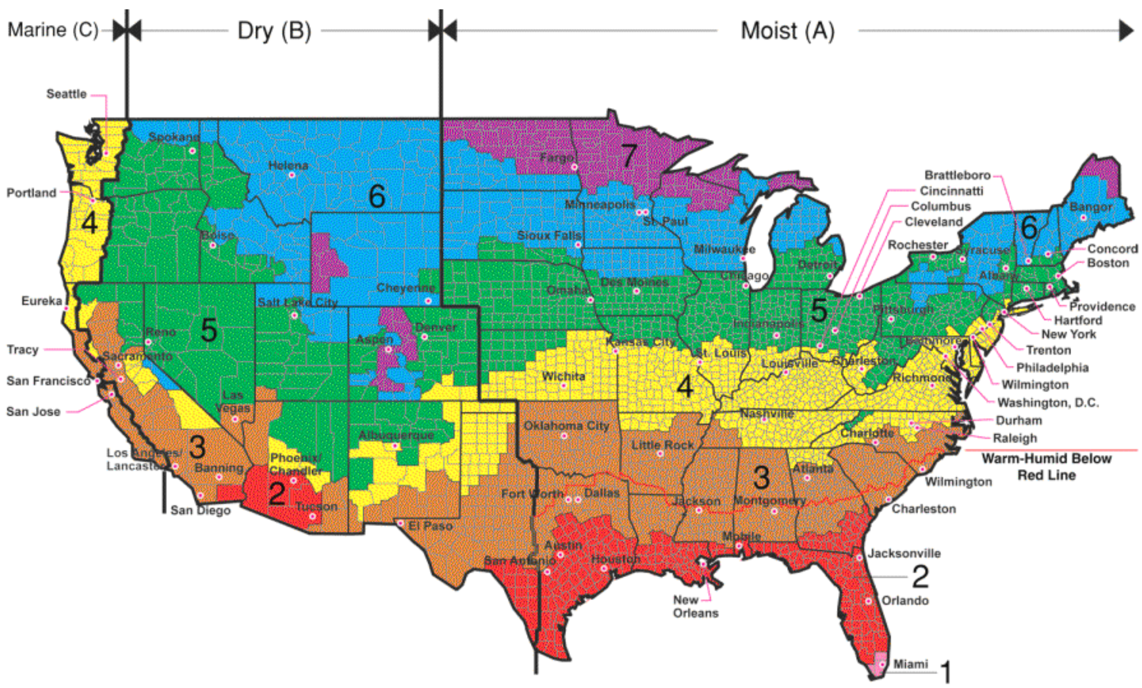


Figure 1.3: Psychrometric chart showing fixed dry bulb, fixed enthalpy, fixed dry bulb and dew point and enthalpy curve

Figure 1.3 shows fixed dry bulb, fixed enthalpy and fixed dry bulb and dew point economizer strategies. Region A is same as in Figure 1.2. A fixed dry bulb control option would choose 100% outdoor air in regions B, F, E and minimum outdoor air in regions C and D. Dampers would allow 100% outdoor air for a fixed enthalpy control in regions B, C and F and the damper would be closed if outdoor air lies in the region D and E. Region B represent the region when a fixed dew point and dry bulb economizer strategy will open the damper. The enthalpy curve is also shown in Figure 1.3. The damper would be in fully open position for outdoor air conditions between the left of the curve and the right boundary line separating region A. Minimum outdoor air would be allowed for rest of the outdoor conditions on the right of the curve for this economizer strategy.

1.2 Applicability to climatic zones

Different economizer strategies are applied in different climatic conditions to realize the energy savings. All the economizer strategies should reduce the outside air intake to minimum when no energy savings could be achieved by using outside air. This leads to characterization of control options in different climatic zones. Figure 1.4 shows the distribution of climatic zones in USA. Table 1.2 represents the recommended ASHRAE control strategies for different climatic zones in USA.



Source:www.buildingscienceconsulting.com/designthatwork/images/climate_zones.gif

Figure 1.4: Climatic zones in USA

Table 1.2: Control strategies recommended by ASHRAE 90.1-2007 for different climates

Climate Zones	Allowed Control Options	Prohibited Options
1b,2b,3b,3c,4b,4c,5b, 5c,6b,7,8	Fixed Dry Bulb Differential Enthalpy Differential Dry Bulb Electronic Enthalpy Dew Point and Dry Bulb Temperatures	Fixed Enthalpy
1a,2a,3a,4a	Fixed Dry Bulb Differential Enthalpy Fixed Enthalpy Electronic Enthalpy Dew Point and Dry Bulb Temperatures	Differential Dry Bulb
All Other Climates	Fixed Dry Bulb Fixed Enthalpy Differential Enthalpy Differential Dry Bulb Electronic Enthalpy Dew Point and Dry Bulb Temperatures	

1.3 Model based economizer strategies

Model based economizer strategies use a HVAC system simulation to estimate the cooling coil load at different fractions of outdoor air. The damper position is controlled in a way to minimize the cooling load on the coil. The main purpose of the current study is to evaluate potential energy savings of new model based strategies. The second objective is to evaluate the impact of economizer selection on system design. The study accomplishes this by the following procedure:

1. Previous works are examined to identify what makes different economizer strategies perform better in different climatic conditions.
2. The model based economizer strategy is developed and implemented in EnergyPlus.
3. The EnergyPlus implementation is verified by comparing simulation results with previously reported work.

4. Annual cooling coil energy for the conventional and model based economizer strategies is compared for three benchmark buildings and fifteen locations representing all US climatic regions.
5. The impact of economizer strategy on cooling coil design is evaluated.

The thesis consists of 10 chapters, including introduction as chapter one. Chapter two contains the literature review to recognize the previous efforts done evaluating economizer strategies. Chapter three describes the methodology developed for the new model based economizer strategies. Chapter four discusses the implementation of the model based control strategies in EnergyPlus. Chapter five deals with the verification of economizer strategies as they are implemented in EnergyPlus. Chapter six is a brief introduction of the different types of buildings chosen for parametric study and their importance. Chapter seven discusses the results for annual cooling coil energy with different economizer strategies for perfect and imperfect sensors for the three building types. Chapter eight then highlights the impact on design of equipment due to economizer selection. Chapter nine is a summary of all the results and discussion about them. Some recommendations are made based on the analysis of economizer strategies for designers using simulation codes. The last chapter is the list of references followed by a set of appendices.

CHAPTER 2

2. Literature Review

Temperature based and enthalpy based economy cycles are the most common economizer strategies. Several different researchers have tried to compare the energy savings associated with using these different economizer strategies in different climatic conditions.

Hittle and Johnson (1985) pointed out problems with humidity sensors which are used in measuring humidity while using differential enthalpy economizer strategy. This makes it difficult to rely on this strategy for the energy savings. Kao and Pierce (1983) demonstrated that a moderate sensor error can cause significant increase in system energy consumptions. Sometimes this increase would negate the effect of savings associated with using differential enthalpy cycle.

Spitler (1987) and others studied two buildings with differential dry bulb and differential enthalpy cycle in five different locations. The results showed that differential enthalpy provides extra energy savings than differential dry bulb almost at every location ranging from 10.2% to -0.44%. However, they presented an observation as shown in Table 2.1 that enthalpy based economizer could still result in higher cooling coil load than temperature based control. This is due to the increase in the sensible cooling load on the coil when using 100% outdoor air with enthalpy based control. Although, it was found that the resulting relative humidity of supply air is high when using temperature based control. Thus, enthalpy based control would still have an advantage when humidity is also to be controlled. They also concluded that the energy savings achieved through use of enthalpy based economizer is marginally higher than that of temperature based control in dry climates. But realizing these savings would not be practical considering the unreliability of humidity sensors, they recommended using temperature based economy cycles in those climates. However, approximately 16% savings were obtained in Houston but given the unpredictable nature of humidity sensors, it was recommended using no economizer in hot and humid climates unless the additional controls could be cost effective.

Table 2.1: Comparison of economizer cycles

	Return Air	Outdoor Air
Dry Bulb Temperature(C)	27.2	29.4
Specific Humidity	0.00880	0.00510
Relative Humidity (%)	40	20
Enthalpy (kJ/kg)	67.8	60.9
	Flow over Cooling coil	
	Differential Enthalpy Cycle	Differential Dry Bulb
Fraction Outdoor air	1.0	0.2
Entering Conditions		
Inlet Dry-Bulb (C)	29.4	27.6
Specific Humidity	0.00510	0.00806
Relative Humidity (%)	20	36
Enthalpy (kJ/kg)	60.9	66.4
Exit Conditions		
Inlet Dry-Bulb (C)	15	15
Specific Humidity	0.00510	0.00806
Relative Humidity (%)	48	76
Enthalpy (kJ/kg)	46.2	53.9
Enthalpy Difference(kJ/kg)	14.7	12.5

Wacker (1989) did an economizer savings study based on indoor comfort levels in a commercial building in six cities. He deduced through his simulations that differential dry bulb would save some energy over differential enthalpy but lose the space comfort level. He concluded that differential enthalpy would give the most energy savings without discomfort.

Khutoryanskiy and Margadant (1999) investigated another method of reducing the load and energy usage in a commercial building in San Francisco. They found that the differences in the air temperature on the adjacent or opposite sides of buildings can be as much as 10-12 F during a day. This meant that air intake at one side is not the best possible choice. They concluded that an economizer should also have a choice to the side of outdoor air intake. However, this work did not involve improving or comparing any economizer strategy.

Yiu(2000) found the effectiveness of enthalpy based economizer in humid climate of Hong Kong. He studied the use of differential enthalpy economizer cycle in urban, suburban and rural place. He concluded that over 50% of time in winter, economizers could be used for free or partial cooling. Engineers in Hong Kong did not use economizer controls and feasibility of these controls were not evaluated before. Through his practical experiments, Yiu concluded that using an economizer would actually be very effective in climatic conditions like that of Hong Kong.

Budaiwi(2001) compared the energy usage under three climatic conditions in Saudi Arabia in a nine story office building . The three places chosen represented hot-humid summer and mild winter in Dhahran, moderate summer and winter in Khamis Mushait and hot-dry summer and cold winter in Tabuk. Cooling coil energy was measured with and without the use of economizer. Two economizer strategies viz. differential dry bulb and differential enthalpy were tested. He observed that there were potential energy savings using economizers in these climatic conditions. Highest savings were achieved in hot dry summer and cold winter, moderate savings in moderate summer and winter and least savings in hot humid summer and mild winter. He also concluded that differential enthalpy had no advantage over differential dry bulb for high humidity controls. He also stated that increasing the supply temperature could increase in the savings.

All these studies have confirmed that economizers could potentially save lot of energy when installed in a building system. Some studies have also stated that it is not advisable to use economizers in hot and humid climates (Spitler 1987). However, other studies have shown a little savings in that area (Budaiwi, 2001). Moreover, all the studies mainly focused on comparing energy savings using temperature based and enthalpy based economizer strategies. Temperature based strategies was found to have advantage in some areas and enthalpy based in others.

In search of a new economizer modeling strategy that could be used under any climatic conditions, Seem and House (2009) proposed new control strategies for controlling air economizers. The new economizer strategies are called as model based strategies as they use a model for estimation of cooling load on the cooling coil and adjusting the outside air dampers in a way to have minimum cooling coil load. They used a bypass cooling coil model for cooling load calculation considering a constant air volume system using MATLAB as simulation platform. The coil model is an approximation of actual system coil model assuming coil to be completely dry or completely wet. The return air conditions and supply air temperature were assumed to be fixed and the testing was done over 15 different cities across US. Sensor errors were also implemented to evaluate the sensitivity of peak loads and cooling coil energy. A similar approach is adapted to implement the new modeling strategies in EnergyPlus. One of the important things is to eliminate the assumptions of Seem and House (2009) study and represent results for real building simulations. The coming chapters describe the development and implementation of model based strategies in EnergyPlus.

The next chapter explains the methodology used to develop the new model based economizer strategies. Chapter 4 explains the implementation of the methodology in EnergyPlus. The chapter also describes the two different coil models used to calculate cooling coil loads and strikes the difference between the two approaches. Chapter 5 deals with evaluation of the economizer strategies and verifies the results obtained. Sensor errors and their affect on control decision and thus the effect on energy savings are followed up in another discussion. A parametric study is done with three different building types for various locations across US. The building types are explained in chapter 6. The savings in cooling coil energy are compared for different economizer strategies in chapter 7 while the impact on design based on annual peak cooling loads is discussed in chapter 8.

CHAPTER 3

3. Developing Model Based Economizer Strategies

The new economizer strategies are called model based economizer strategies as they use a coil model to estimate the load on the cooling coil and thus controlling the outside air and return air dampers to minimize the cooling load. As presented by Seem and House (2009), Figure 3.1 shows the psychrometric chart divided into regions based on the mechanical cooling load. Region A or the white colored portion represents outdoor air conditions that could provide free cooling. The light grey portion represents the region where 100% outdoor air would give minimum cooling load and dark grey portion represents portion where minimum outdoor air would give minimum cooling load. Point P represents the return air conditions.

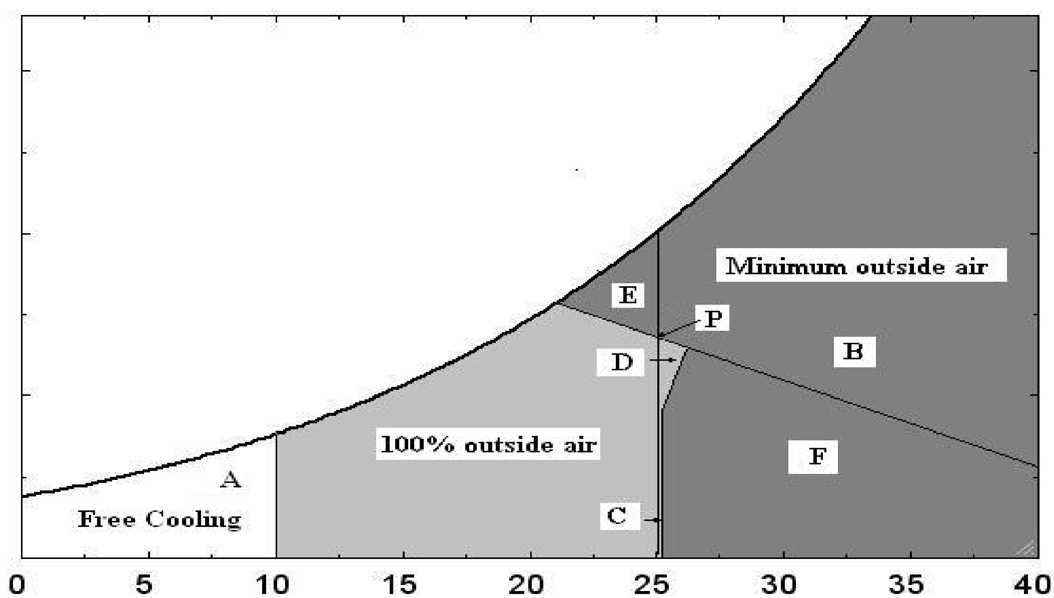


Figure 3.1: Psychrometric chart showing ideal outside flow rate required to minimize cooling coil load (Seem and House, 2009)

The lines passing through point P represent the ideal transition lines. Region E is bounded by constant dry bulb temperature line and saturation line. The outdoor air has high moisture content in this region thus bringing excessive outdoor air would increase the latent load on the cooling coil. In this region differential dry bulb would use 100% outdoor air in this region and thus would waste a lot of energy.

Region F is bounded by return air enthalpy line and the bottom of the psychrometric chart. The outdoor air is warmer and drier than the return air. The humidity of outdoor air for most of the part of region F is low enough so that no condensate forms on the coil. However, moving more right in region F would increase the sensible cooling coil energy. Differential enthalpy in such a case would use 100% outdoor air that would increase the cooling load on the coil.

On the upper left corner of region F, in region D, 100% outdoor air would increase the sensible load on the cooling coil but it reduces the latent load by more than that. A differential dry bulb control would use minimum outdoor air in that case and thus would result in increase in cooling coil energy, but a differential enthalpy control would correctly select 100% outside air.

In region C, outdoor air has slightly higher temperature than return air. As the temperature of both the air streams is almost the same, the temperature of mixed air is approximately constant regardless of the amount of outdoor air flow rate. Differential dry bulb would select minimum outside air in this case which would mean that the mixed air would have slightly higher humidity and thus slightly higher enthalpy content. As the temperature of mixed air is same, the increase in its enthalpy would mean increase in sensible cooling energy required for cooling. Thus, the correct choice would be to operate with 100% outdoor air.

As shown, both differential temperature and differential enthalpy economizer strategies make bad decisions in different regions of the psychrometric chart. Model based strategies are developed to avoid these bad decisions. These strategies use the cooling load on the cooling coil as the controlling factor for damper positions rather than just comparing outdoor and return air conditions.

3.1 Model Based Control Strategies

As stated earlier that model based control strategies are based on adjusting the damper position at outdoor air intake so as to reduce the mechanical cooling required. The model based controls are classified in two types based on damper position and coil model used for damper control.

Classification based on outside air damper position:

1. Two position: The outside air damper is controlled to allow either minimum outside air with damper being closed (minimum position) or 100% outdoor air with fully open damper.
2. Optimal position: The outside air damper can be controlled to have any damper position between closed and fully open to allow the respective fraction of outdoor air to realize more energy savings associated with other damper positions than at minimum and maximum positions.

Classification based on coil models for damper control:

1. Bypass coil model: This is a simple cooling coil model that assumes a fraction of air bypasses the cooling coil and the remaining fraction comes in contact with coil. The coil is either considered dry or fully wet.

2. Detailed coil model: This cooling coil model does not assume any bypassed air and actually determines if the coil is dry, wet or partially wet.

3.1.1 Two position model based control strategy

The cooling loads on the coil are calculated for two positions of damper viz. fully open and closed conditions. The mixed air is obtained in such a way that it either has minimum fraction of outdoor air or is 100% outdoor air. The cooling loads are obtained on the cooling coil using the mixed air conditions as the air side inlet conditions.

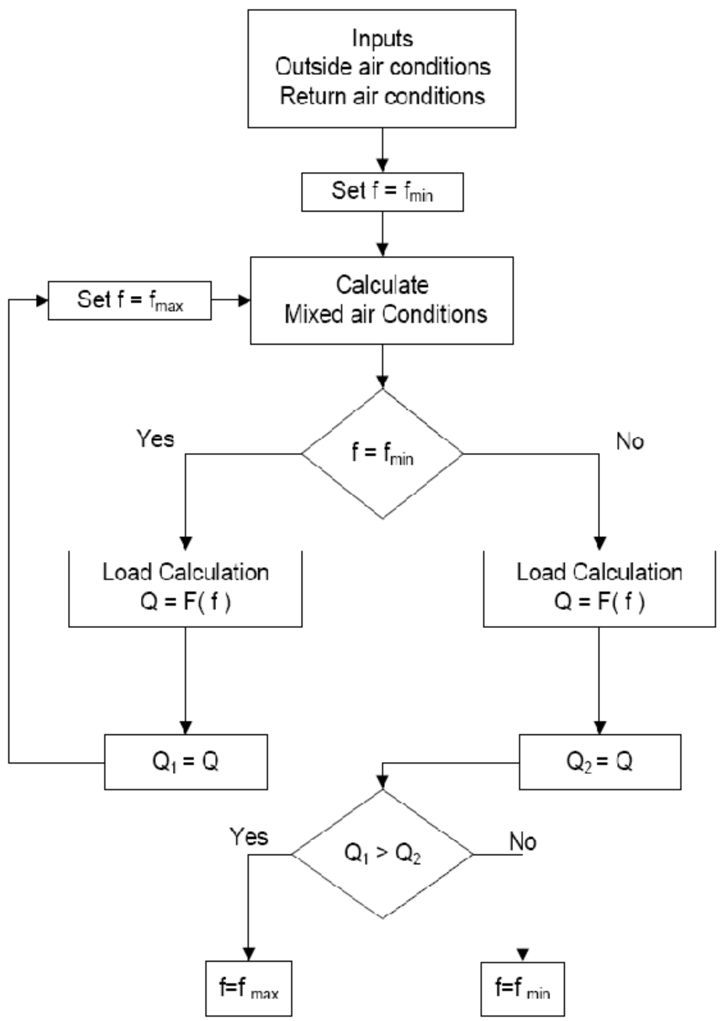


Figure 3.2: Flow chart for two position model based economizer strategy

The cooling loads are obtained for both the mixed air conditions. The cooling loads on the coil are then compared against each other. A decision is made to control the damper position at minimum or 100% outdoor air depending upon which load is less. The process is represented in Figure 3.2.

3.1.2 Optimal position model based control strategy

The optimal position model based control strategy for economizers allows the modulation of damper position between the closed and fully open position. Figure 3.3 represents the algorithm for this. The algorithm follows an optimization routine that uses a golden section search method to determine the fraction between minimum and maximum possible that would give the least cooling load on coil based on golden section method. The golden section number tau is given as $\frac{\sqrt{5}-1}{2}$ or 0.618 approximately. The fractions are then used to calculate the mixed air conditions and then the load on the cooling coil. The loads are then compared to determine the minimum of two. The range of fractions is continuously reduced following an iterative procedure towards the fraction giving the least load. When the range becomes less than a very small number, the optimization is assumed to be complete and the damper position at outside air inlet is adjusted to the converged outside air fraction.

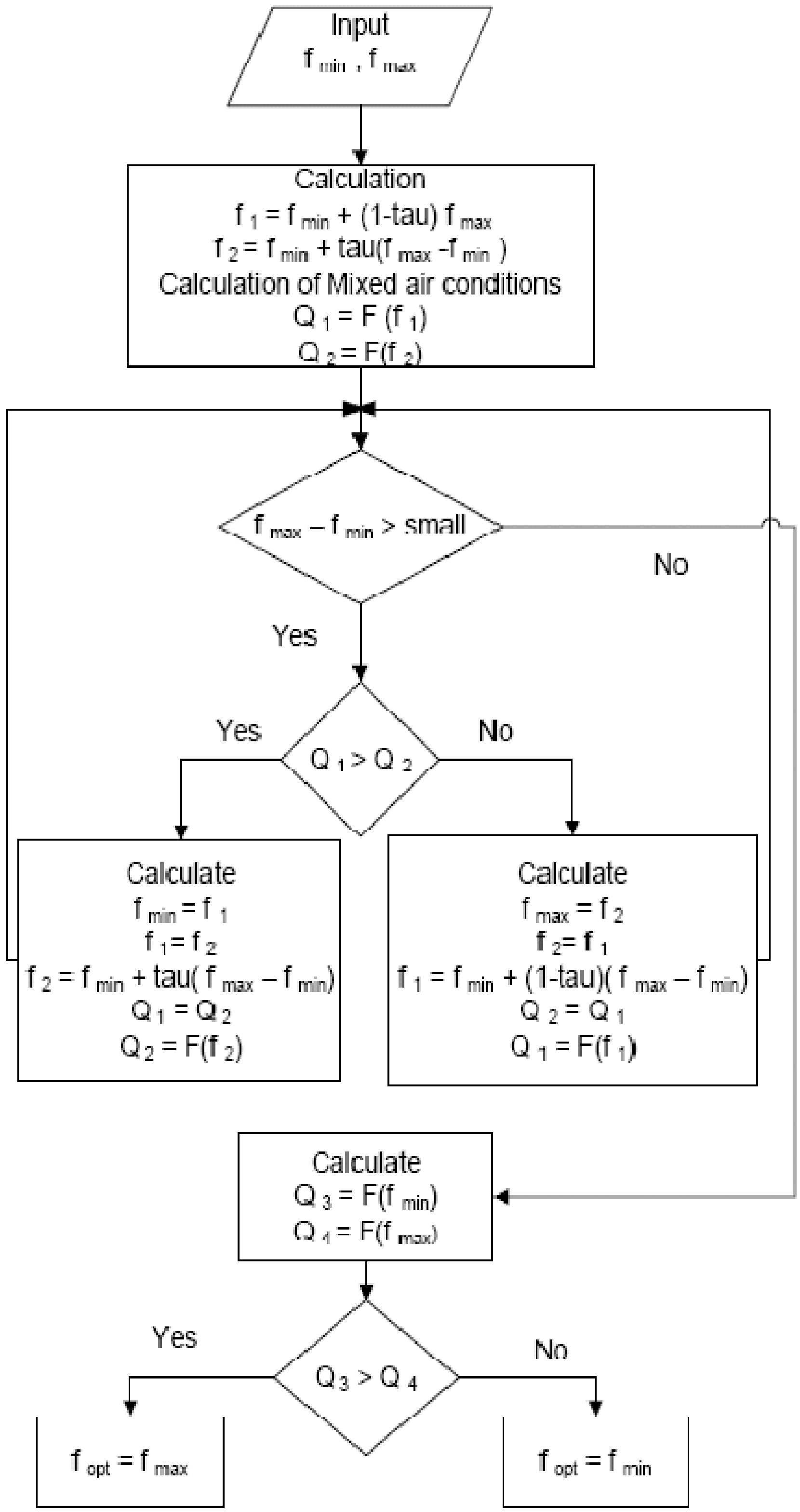


Figure 3.3: Flow chart for optimal position model based economizer strategy

The methodology is straightforward but requires understanding of air loop and system components to practically realize them in EnergyPlus. The next chapter discusses the implementation of these strategies in EnergyPlus along with the conventional economizer strategies already present. A description of both detailed and bypass coil models is also given.

CHAPTER 4

4. Implementing Model Based Economizer Strategies in EnergyPlus

Conventional economizer strategies control the damper position based on the comparison of outside air and return air conditions. However, in model based controls, the determination of damper position is based on load calculation on coil for different fractions of outside air. To implement these strategies in EnergyPlus, it becomes necessary to understand the system setup.

4.1 System Description

The arrangement of the air loop could be of two types depending upon the position of fan. A draw through system is shown in Figure 4.1 or a blow through system which is shown in Figure 4.2 .

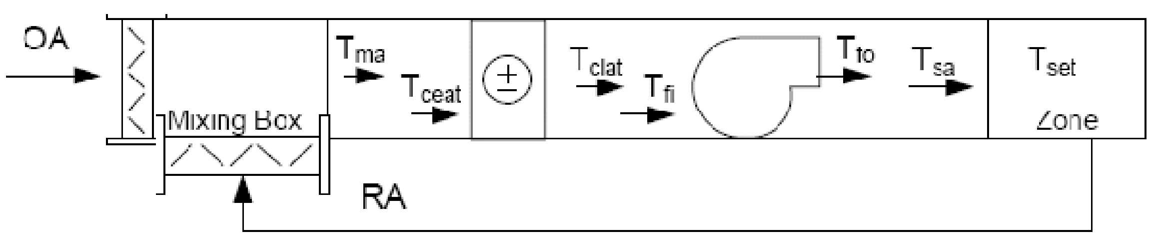


Figure 4.1: Draw through system

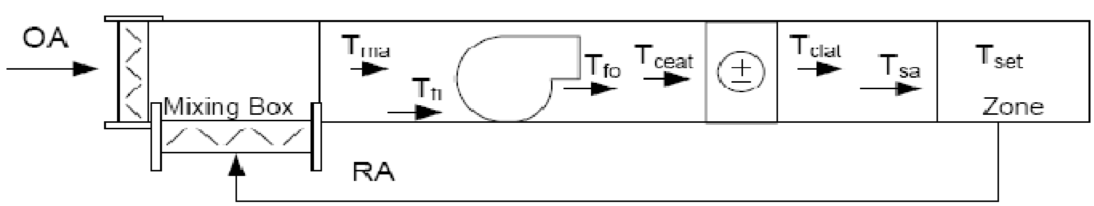


Figure 4.2: Blow through system

Outside air dampers are controlled by an economizer strategy. The mixing of outdoor air and return air takes place in the mixing box. In a simulation, the enthalpy and humidity ratio are calculated based on the fractions of outdoor air and return air. Temperature of mixed air is then calculated using the psychrometric functions.

$$\omega_{ma} = f\omega_{oa} + (1-f)\omega_{ra} \tag{4.1}$$

$$h_{ma} = fh_{oa} + (1-f)h_{ra} \tag{4.2}$$

To account for heat added by fan and thus the increase in temperature of air across it, it is important in a simulation to determine the coil leaving air temperature (T_{clat}). The cooling coil entering air temperature (T_{ceat}) also depends on the fan arrangement.

For a draw through fan arrangement:

$$T_{clat} = T_{set} - (T_{fo} - T_{fi})\ ,\ T_{ceat} = T_{ma} \tag{4.3}$$

and for a blow through fan arrangement:

$$T_{clat} = T_{set}\ ,\ \ T_{ceat} = T_{fo} \tag{4.4}$$

To take care of this increase in temperature due to fan heat, a controlled mixed set point temperature (T_{mst}) is calculated in the simulation. This is always given as :

$$T_{mst} = T_{set} - (T_{fo} - T_{fi}) \tag{4.5}$$

Calculation of this is important as it determines the conditions when free cooling is possible. If the cooling coil has enough capacity then supply air temperature (T_{sa}) should always be equal to set point temperature (T_{set}).

4.2 Implementing conventional strategies in EnergyPlus

EnergyPlus is a simulation program that models heating, cooling, lighting, ventilating, energy and water flow in a building system. Figure 4.3 presents an overview of EnergyPlus.

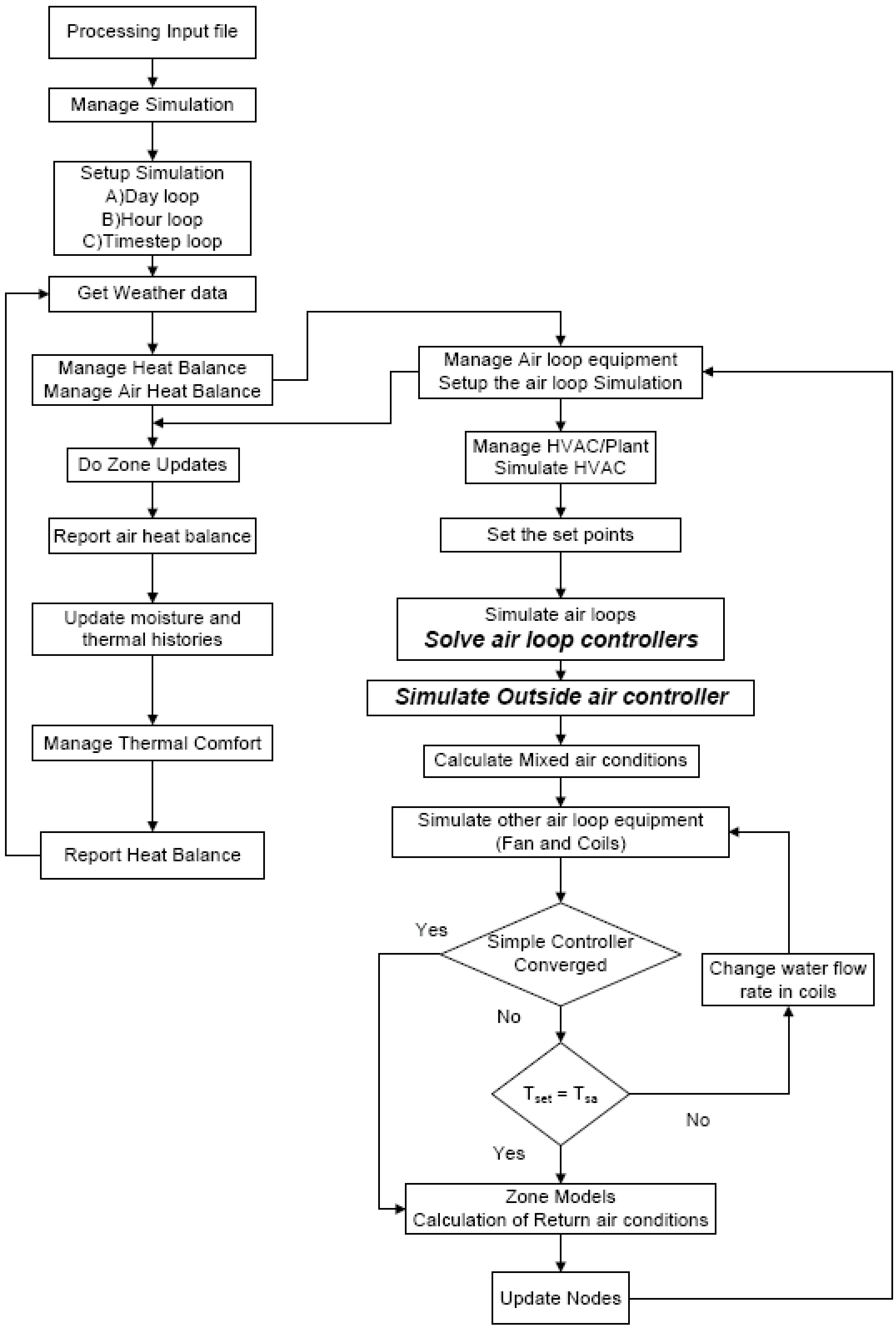


Figure 4.3: EnergyPlus simulation flow

The whole simulation is controlled by a simulation manager in EnergyPlus. Energy and mass balances equations are combined with steady state component models. When the zone air system is modeled, there arise one or more algebraic loops due to the equations. EnergyPlus decouples the zone calculations from air system equations. Zone air conditions are predicted and kept constant throughout the system simulation. The system simulation results are then used to correct the predicted zone values. This is done till a system convergence is reached.

The air loop equipment is simulated in the order they are connected to each other. The air loop consists of outside air controller connected with the coils (cooling and heating) and fans. The air is then distributed to individual zones through supply ducts. The cooling coils and heating coils are controlled by a simple controller which controls the flow of water through these coils iteratively to bring air temperature equal to the controlled temperature. The nodes are updated and the simulation continues. The conventional economizer strategies are present in outside air controller. The damper position is calculated using these economizer strategies and is then used to calculate mixed air conditions.

4.2.1 Calculation of Damper position

The first step in air loop simulation is calculation of damper position. Conventional economizer strategies are placed in outside air controller for controlling the damper positions as shown in Figure 4.4. If no economizer is used then the damper would always be at closed (minimum) position. Except for no economizer, every other economizer strategy should provide free cooling whenever it’s possible. If an economizer is used and the outside air temperature is more than the mixed set temperature, the fraction of outside air is first set to 1 and then the economizer strategies are checked to determine if any of those sets the fraction to minimum.

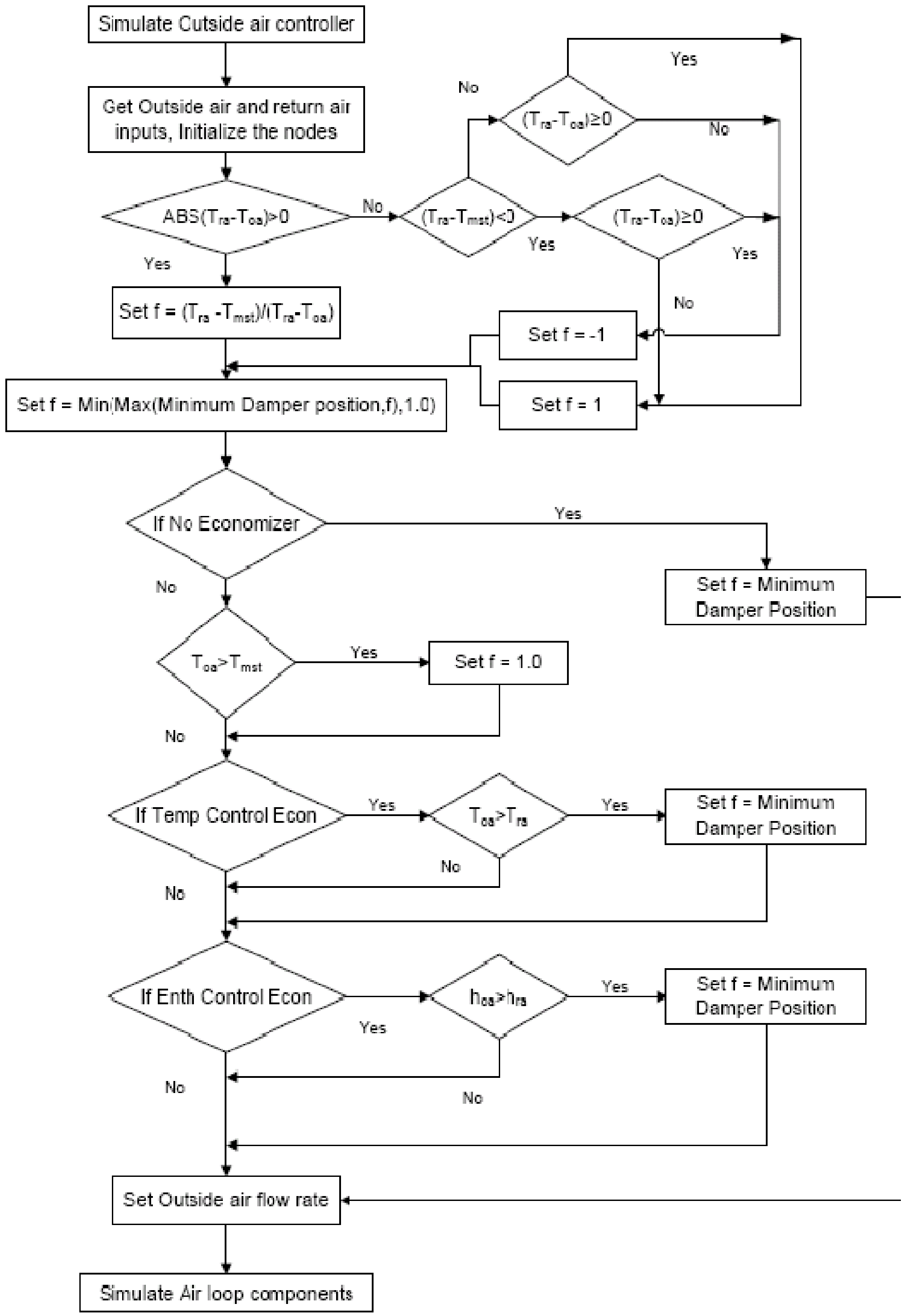


Figure 4.4: Conventional strategies in outside air controller

The simulation in EnergyPlus is setup in such a way that it seems that several strategies can be checked in a single run as shown in Figure 4.4. However, only one economizer strategy could be given as input and thus only one strategy would be active for the simulation. The conventional economizer control strategies would reset the fraction of outside air to minimum if they are found true as shown in Figure 4.4. For example, a differential dry bulb strategy would reset the fraction of outdoor air to minimum if outside air temperature is more than the return air temperature. After the control decision has been made, the signal is transferred to the damper and the damper is adjusted to the closed (minimum outdoor air) or open position (100% outside air) position. Depending on damper position, the outside and recirculated air flow rates are calculated.

4.3 Implementing Model Based Strategies in EnergyPlus

Model based controls were implemented in two ways in EnergyPlus depending upon the type of coil model that was used. The detailed coil based strategy uses the system coil model in EnergyPlus to determine the cooling load and thus control the damper position. The bypass coil based strategy uses the new bypass coil model that was implemented in the controller to control the damper position. However, after a control decision has been made using the bypass coil model, the system (detailed) coil model is used for the actual load calculation. For simulation and modeling purposes, a bypass coil model saves a lot of computation time. Moreover, it is more practical to implement the bypass coil model in the controller since it requires fewer inputs and uses simpler algorithm than the detailed coil model. The coil models and their implementation in EnergyPlus are discussed in coming sections.

4.3.1 Implementing the bypass coil model

The bypass coil model is placed in outside air controller itself as shown in Figure 4.5.

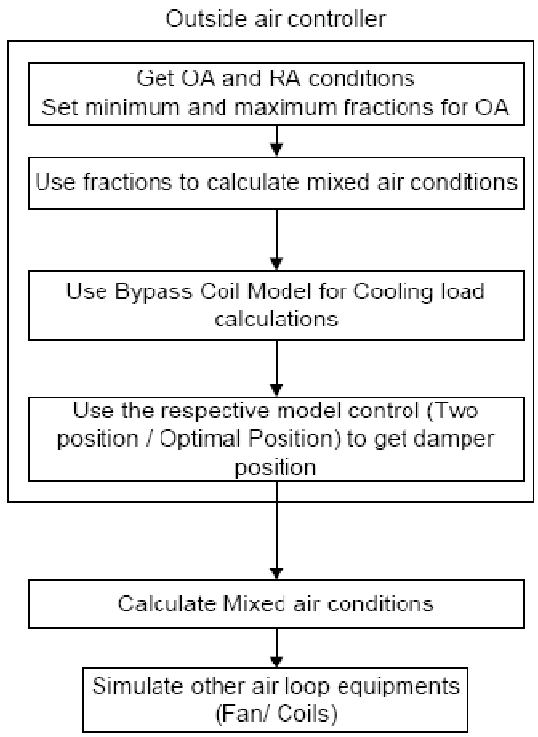


Figure 4.5: Bypass coil controlled model

The bypass cooling coil model is based on contact mixture analogy (Carrier et. al, 1937). The assumption here is that some fraction of the air that enters the cooling coil does not come in contact with the coil and is therefore considered as bypassed as shown in Figure 4.7. The remaining fraction comes in contact with the cooling coil. The bypassed air for that reason is considered to leave the coil with the entering air conditions. The other assumption is that the coil always has enough capacity to achieve the contact air leaving temperature (T_{la}) necessary to meet the mixed set point temperature (T_{mst}). Thus, the fraction of air that comes in contact with coil is considered to be cooled to the temperature that when it is adiabatically mixed with bypassed air would result in air at the mixed set point temperature. Consider a draw through fan: mixed air obtained from the mixing chamber would serve as inlet air to the cooling coil. Figure 4.6 represent the schematic of the simulation system arrangement in EnergyPlus. The bypass coil model is used to make a decision based on the minimum cooling load as calculated by the coil model in the controller. Once the control decision is made, the damper position is set, and the system is simulated using the detailed system coil model.

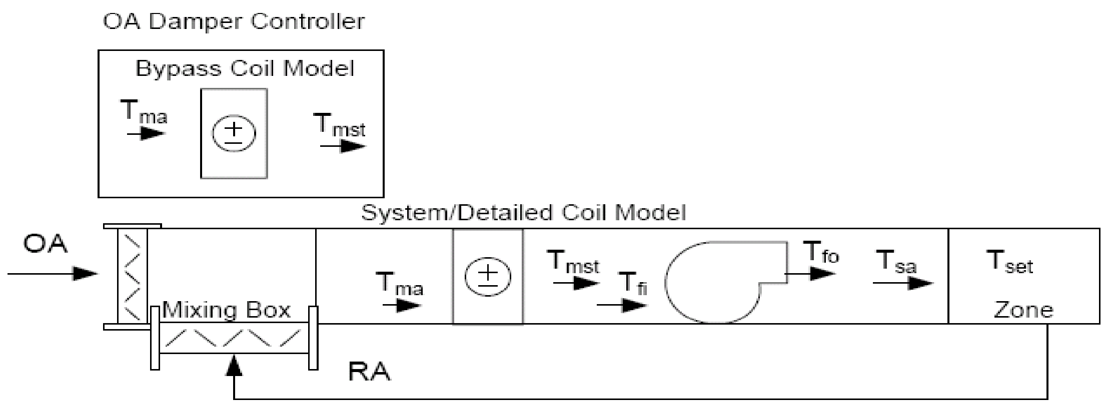


Figure 4.6: Schematic of air loop with bypass coil model used for control decision

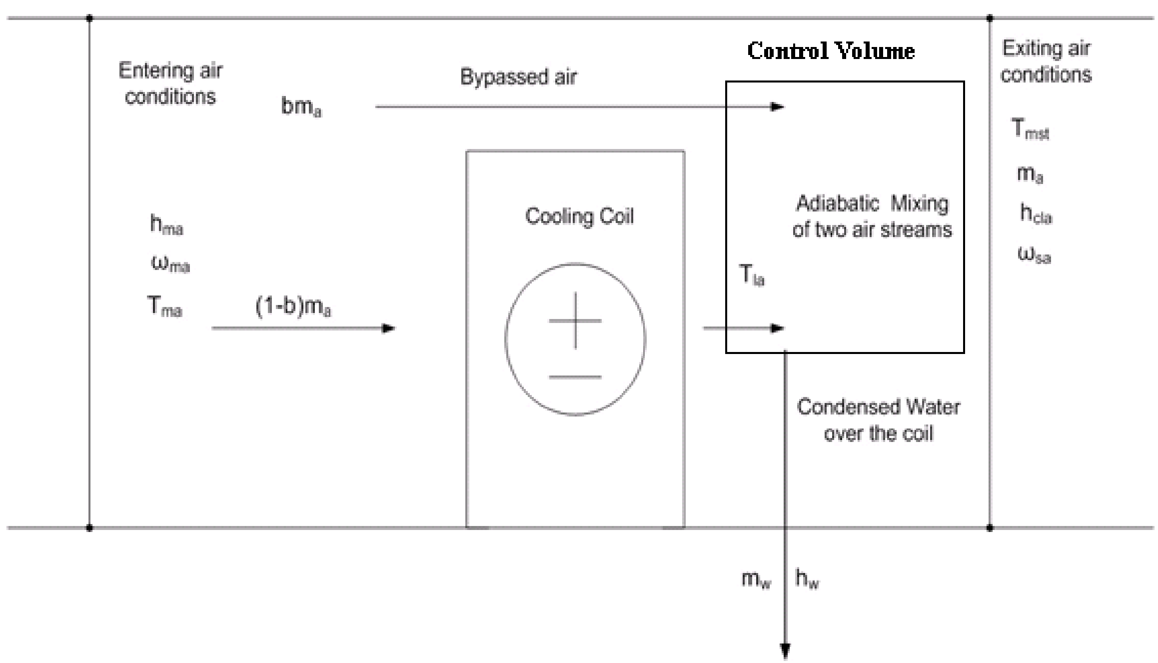


Figure 4.7: Schematic of bypass cooling coil model

One important difference between the detailed coil model and the bypass coil model is how they determine whether the coil is wet or dry. The bypass coil model makes the determination by calculating a transition temperature. The transition temperature is the coil leaving air temperature at which the coil surface transitions from dry to wet. However, there is no condensation at this temperature. Now, if there is no condensation, the leaving air temperature should be the transition temperature. A simple energy balance is applied on the air and water vapor entering and leaving the control volume as shown in Figure 4.7 to obtain transition temperature. If the bypass fraction is given by b then,

$$\mathrm{T}_{ta} = bT_{ma} + (1-b)T_{la} \tag{4.6}$$

T_{la} is the leaving air temperature of the part of air that comes in contact with cooling coil If the coil is in transition or wet, the air that comes in contact with coil is saturated and the temperature of the contact air is equal to the dew point temperature of the air entering the coil. Therefore, the transition temperature can also be written as:

$$\mathrm{T}_{ta} = bT_{ma} + (1-b)T_{dp,ma} \tag{4.7}$$

The mixed set point temperature which should be the actual leaving air temperature is then compared with the transition temperature. The coil is considered dry if mixed set point temperature is more than the transition temperature and wet if it's less. The calculation of transition temperature thus depends on the value of bypass factor. Depending upon how much air is considered as bypassed, the exit conditions of the contact air changes and thus the calculation of humidity ratio of the supply air changes.

The leaving air temperature for the contact air is given by:

$$T_{la} = \frac{T_{mst} - bT_{ma}}{1-b} \tag{4.8}$$

If the coil is dry then the humidity ratio of supply air is same as that of mixed air.

$$\omega_{sa} = \omega_{ma} \quad (4.9)$$

But if the coil is wet then the contact air is considered to be saturated with 100% relative humidity and the humidity ratio of leaving air that comes in contact with cooling coil is calculated likewise. The humidity ratio of supply air is then calculated using Equation 4.10.

$$\omega_{sa} = b * \omega_{ma} + (1 - b)\omega_{la} \quad (4.10)$$

The enthalpy of the leaving air (h_{cla}) is then calculated using the mixed set temperature and supply air humidity ratio.

The load on the cooling coil is then calculated as

$$\frac{Q_{coil}}{m_a} = h_{ma} - h_{cla} - (\omega_{ma} - \omega_{sa})h_w \quad (4.11)$$

The cooling loads are calculated based on different fractions of outdoor air and the modeling strategy is used to determine the fraction of outdoor air that gives the least load.

4.3.2 Implementing the detailed coil model

The detailed coil model is essentially the system coil model in EnergyPlus used for calculating actual load on the cooling coil. Since, this model requires lots of input data which are not available at controller level, the model was controlled from a higher level than outside air controller. The fraction of outdoor air is found using the model based strategy in air loop controller. The fraction is passed to outside air controller and air loop simulation is done to find the load on the coil. This process is performed iteratively until an optimum fraction of outdoor is found which gives the least load on cooling coil. This optimum fraction is used again for a full air loop simulation and updating nodes. The schematic of simulation of system simulation arrangement is shown in Figure 4.8. The system flow is shown in Figure 4.9

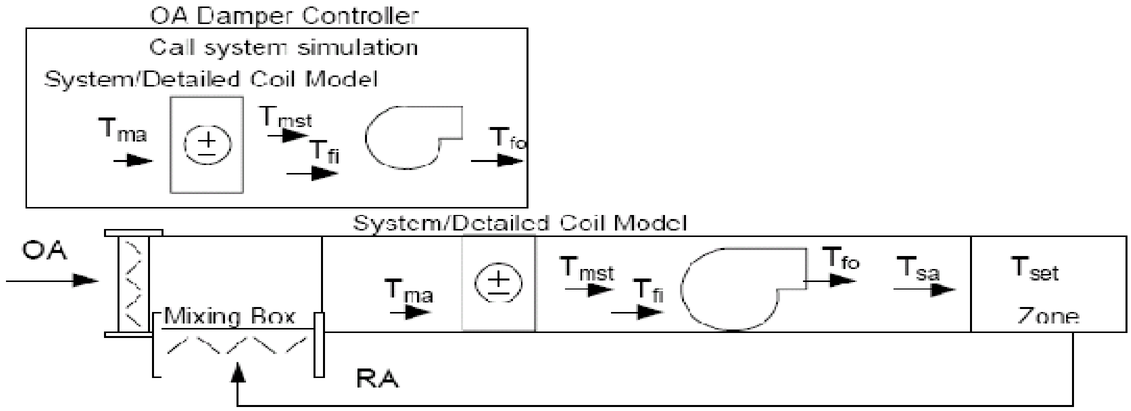


Figure 4.8: Schematic of air loop with system coil model used for control decision

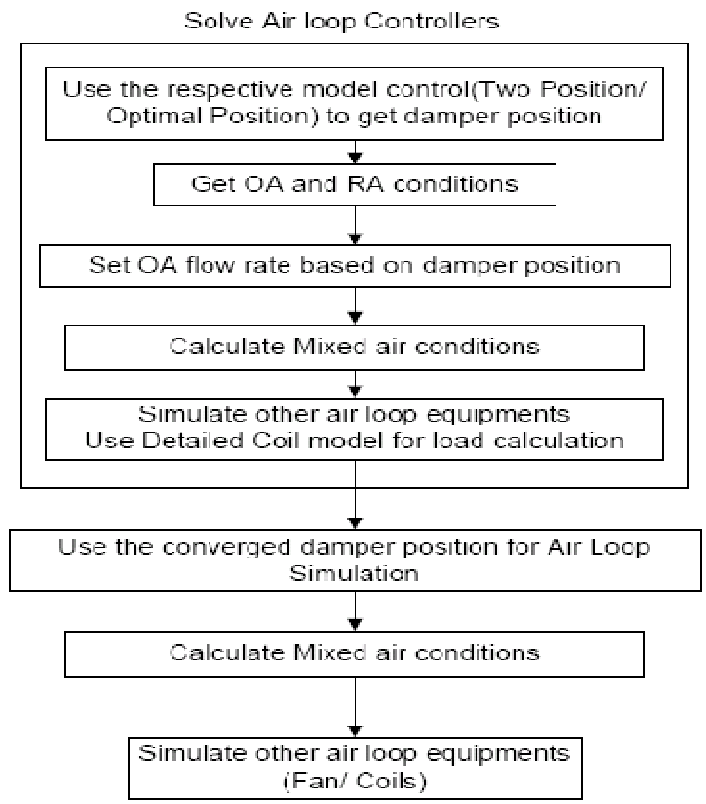


Figure 4.9: Detailed coil controlled model

The detailed coil model in EnergyPlus is implemented using Elmahdy and Mitalas cooling coil model (1977). The coil is considered fully dry if the water inlet temperature is more than the entering air dew point temperature to the coil otherwise it is assumed as completely wet. The assumption lets the model determine surface temperature of the coil at the air inlet assuming that the coil is fully wet. The temperature of coil surface at air inlet is compared against the entering air dew point temperature again. If the entering air dew point temperature is more than coil surface temperature then the assumption of considering coil as completely wet holds true. However, if entering air dew point temperature is less than the coil surface temperature at inlet, then the assumption of a completely wet coil is contradicted. The cooling coil model is then considered to be partially wet. The fraction of the coil that is wet is calculated using equation 4.12.

$$A_{wet}^{coil} = \left(\frac{T_{dp,ma} - T_{wi}}{T_{wo} - T_{wi}}\right) \quad (4.12)$$

The remaining fraction of the cooling coil is considered to be dry and is simulated considering that part as a dry coil. Exiting air conditions are obtained for the dry part of the coil. These exiting conditions are considering as entering conditions for the other part of the coil which is considered as fully wet. Thus, the rest of the coil is simulated as completely wet coil. The loads obtained from the dry part of the coil and the wet part of the coil are added together to give total coil load in this case.

Some of the important equations that are used to calculate the total coil load are given here:

The capacitance of the air is given by

$$C_a = m_a * C_{p,a} \quad (4.13)$$

where $C_{p,a}$ is the constant pressure specific heat of dry air.

The capacitance of water is given by

$$C_w = m_w * C_{p,w} \quad (4.14)$$

where $C_{p,w}$ is the constant pressure specific heat of water.

The cooling coil model uses Effectiveness-NTU (ε-NTU) method for calculation of total load on coil and the exit conditions of air and water.

The coil surface area, flow area and coefficient of heat transfer are the inputs to the model. Other inputs are the entering air and water conditions. Number of transfer units is calculated using the overall heat transfer coefficient and the stream with minimum capacity.

$$NTU = \frac{UA}{C_{min}} \qquad (4.15)$$

where $C_{min} = Min(C_a, C_w)$, is the minimum capacity. (4.16)

Effectiveness (ε) of the cooling coil is then calculated using NTU and ratio of stream capacities (Kays et.al 1964). The maximum heat transfer is possible from the stream that has the minimum capacity. Thus, the maximum heat transfer is calculated as:

$$Q_{max} = C_{min} * (h_{a,in} - h_{w,in}) \qquad (4.17)$$

Exiting enthalpies of the air stream and water stream is calculated using the effectiveness of the coil.

$$h_{a,out} = h_{a,in} - \frac{\varepsilon Q_{max}}{C_a} \qquad (4.18)$$

$$h_{w,out} = h_{w,in} + \frac{\varepsilon Q_{max}}{C_w} \qquad (4.19)$$

If the coil is considered wet then temperature of air at the coil surface is calculated. This temperature is compared to dew point temperature again. If the former is greater than later, then coil is considered partially wet and the exit conditions of air and water are calculated again.

The total load on the coil is calculated as

$$Q_{coil} = m_a * (h_{a,in} - h_{a,out}) \qquad (4.20)$$

The use of detailed coil controller model takes a lot of computation time and is practically very difficult to implement in a real controller.

The next chapter involves the evaluation and verification of the conventional and model based strategies. The verification is done using the results obtained from Seem and House (2009).

CHAPTER 5

5. Model Evaluation and Verification

This chapter describes the evaluation tests and verification tests of conventional economizer and the new model based economizer strategies in EnergyPlus. Model evaluation means that these strategies are making the decision what they should be making according to the specified protocol. The verification tests have been done by comparing the EnergyPlus results with MATLAB results obtained for various economizer strategies. The results from MATLAB are obtained from the bypass mixture model explained earlier. The model in MATLAB was developed by Seem and House (2009).

The economizer strategies are simulated for a large office building model which is used as the input file. Seven different US locations are used for verification purposes. The outside air conditions are obtained from the weather files. The return air temperature is fixed at 25°C and the humidity ratio of return air is obtained by adding a constant 0.0015 to the supply air humidity ratio. The supply air temperature is fixed at 13°C. The inlet temperature of water to the cooling coil is fixed at 5.7 °C. Figure 5.1 shows the schematic of simulation setup in EnergyPlus considering a draw through fan arrangement.

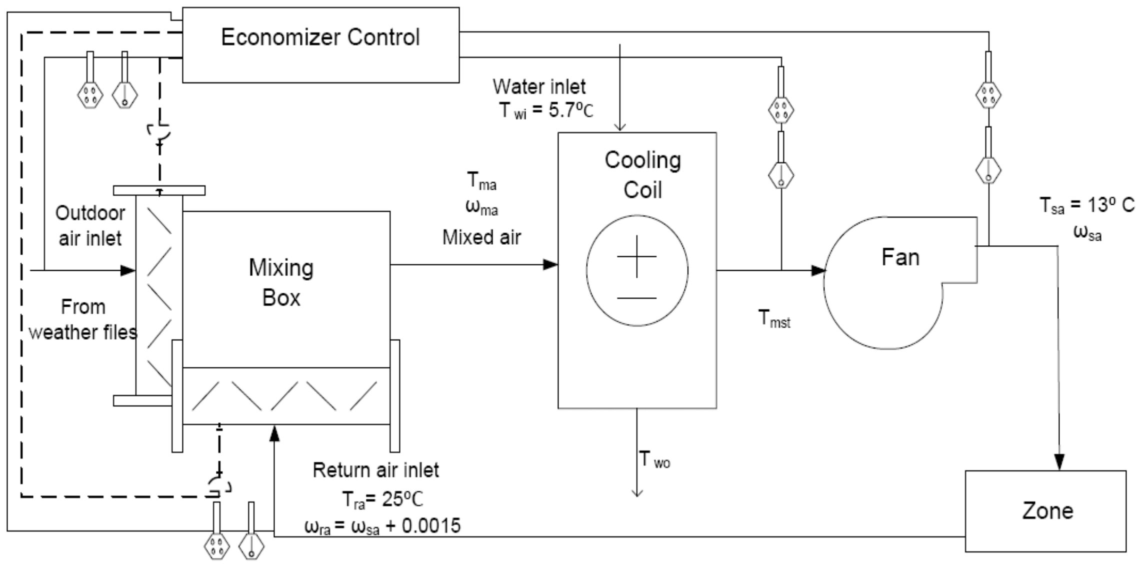


Figure 5.1: Schematic of simulation setup in EnergyPlus

The cooling coil is a cross flow heat exchanger and has enough capacity to meet the cooling coil loads throughout the year. The minimum position of damper corresponds to fraction of outdoor air of 0.2. The system is simulated in EnergyPlus with 1 hour time steps.

Seven different economizer strategies are used for damper control. The strategies are no economizer, differential enthalpy, differential dry bulb, two position bypass coil model, optimal position bypass coil model, two position detailed coil model and optimal position detailed coil model based control strategy.

Hourly averaged output variables are reported for verification. Appropriate outputs are chosen to represent the strategies in the graphs in next section. The output variables include outside air temperature, outside air enthalpy, return air temperature, and return air enthalpy, damper position, and total coil load at minimum fraction, maximum fraction and optimum fraction of outside air. Cumulative cooling coil energy is also reported for comparison with the Seem and House (2009) results.

5.1 Model Evaluation

For performing the evaluation of conventional and model based economizer strategies, two different days were chosen in a year to represent the control decisions made by these strategies for one of the cities (Phoenix). The first day is 18 April which is late spring and early summer and the second is 19 June which represent typical summer day in Phoenix.

The results are plotted for 24 hours of these days with primary y-axis representing temperature, enthalpy or cooling load scale. The secondary y-axis represents the damper position. All the variables plotted on y axis are averaged hourly outputs from EnergyPlus. It is understood that all strategies except optimal position based would choose either minimum or maximum fraction but since the outputs are averaged, it might represent some values between 0.2 and 1. This is due to the fact that EnergyPlus could reduce the system time step to less than an hour for convergence and make different control decision within those time steps. However, the outputs are reported hourly and thus are averaged. The damper is considered closed at minimum fraction (0.2) and fully open at outside air fraction of 1.0. It is also important to understand that the return air conditions except return air temperature may vary in different economizer strategies as per the control decision made by it.

Some of the damper positions as shown in the Figure 5.5 and Figure 5.6 are different from the two control decisions possible being just open or closed. However, in actuality the damper position is controlled to either the closed position (minimum) or the fully open position. The intermediate points shown in the plot are due to the averaging of the short time steps results reported by EnergyPlus. As pointed out earlier, the outputs represented here are the averaged value for an hour.

Some of the results are expressed in coming sections for different economizer strategies.

5.1.1 No economizer

With no economizer strategy, the damper is always closed to allow minimum outside air as shown in Figure 5.2.

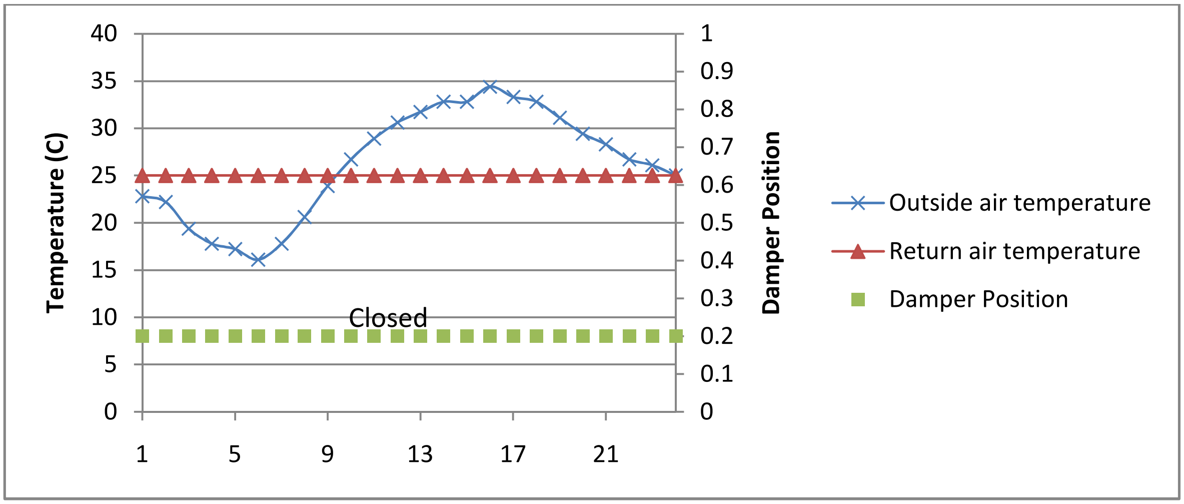


Figure 5.2: Damper always closed with no economizer

5.1.2 Differential dry bulb

Figure 5.3 shows that differential dry bulb control option would open the damper when outside air temperature is less than the return air temperature. The damper would be at closed (minimum fraction) position when outdoor air temperature is higher. Figure 5.4 illustrates the same for 19 June.

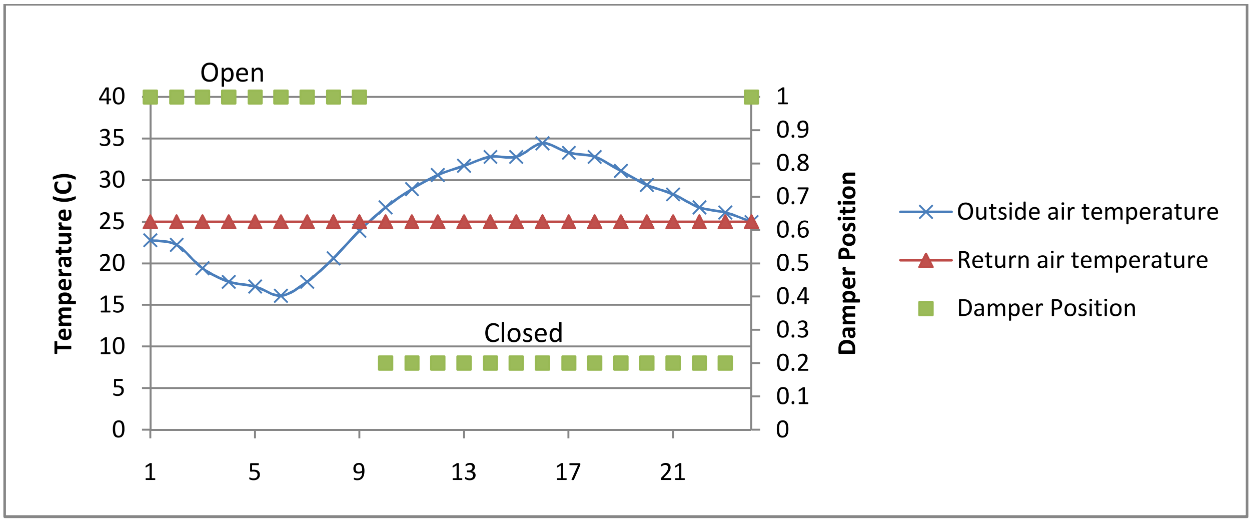


Figure 5.3: Dampers showing open and closed position based on differential dry bulb for April 18

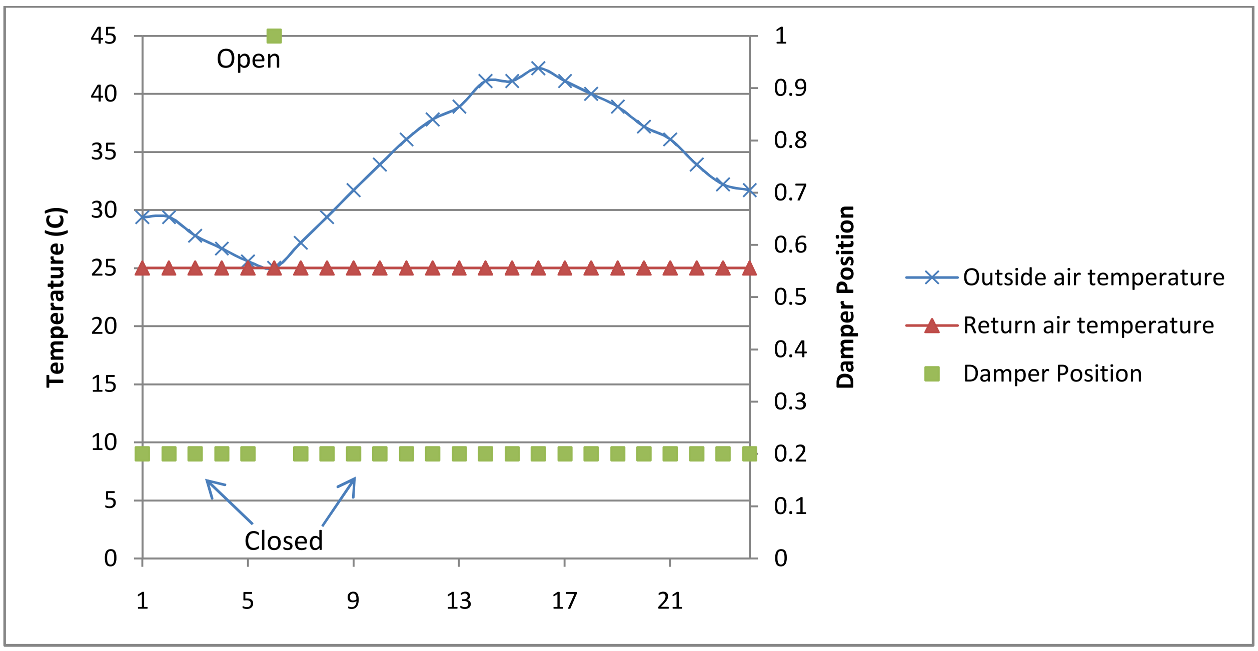


Figure 5.4: Dampers showing open and closed position based on differential dry bulb for June 19

5.1.3 Differential enthalpy

Figure 5.5 and Figure 5.6 show the control decisions for damper position when differential enthalpy economizer is used. The damper is fully open when outdoor air enthalpy is less than the return air enthalpy and is closed when the case is reverse.

Figure 5.5: Damper position with differential enthalpy control strategy for April 18

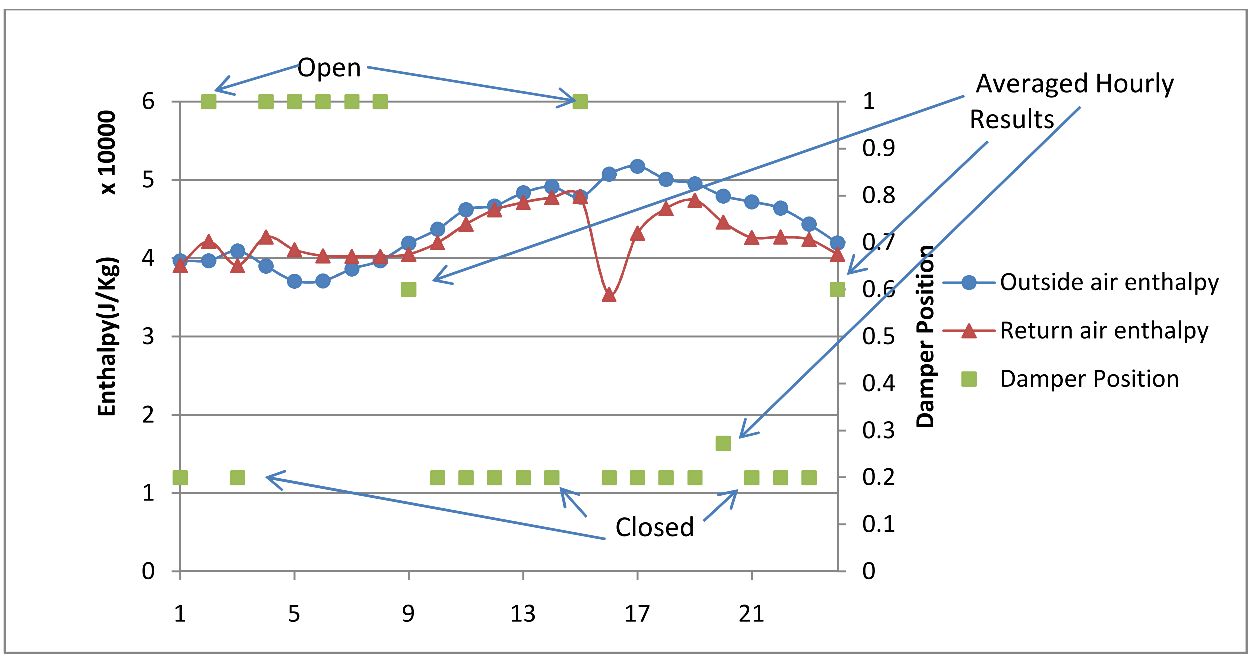


Figure 5.6 : Damper position with differential enthalpy control strategy for June 19

5.1.4 Two position detailed coil model based strategy

Instead of making a control decision based on temperature and enthalpies of outdoor and return air, the Two position detailed coil model based control should adjust damper to position that requires minimum mechanical cooling.

The coil load is calculated using detailed cooling coil model explained in the earlier section. Figure 5.7 and Figure 5.8 show the control decision made by this strategy.

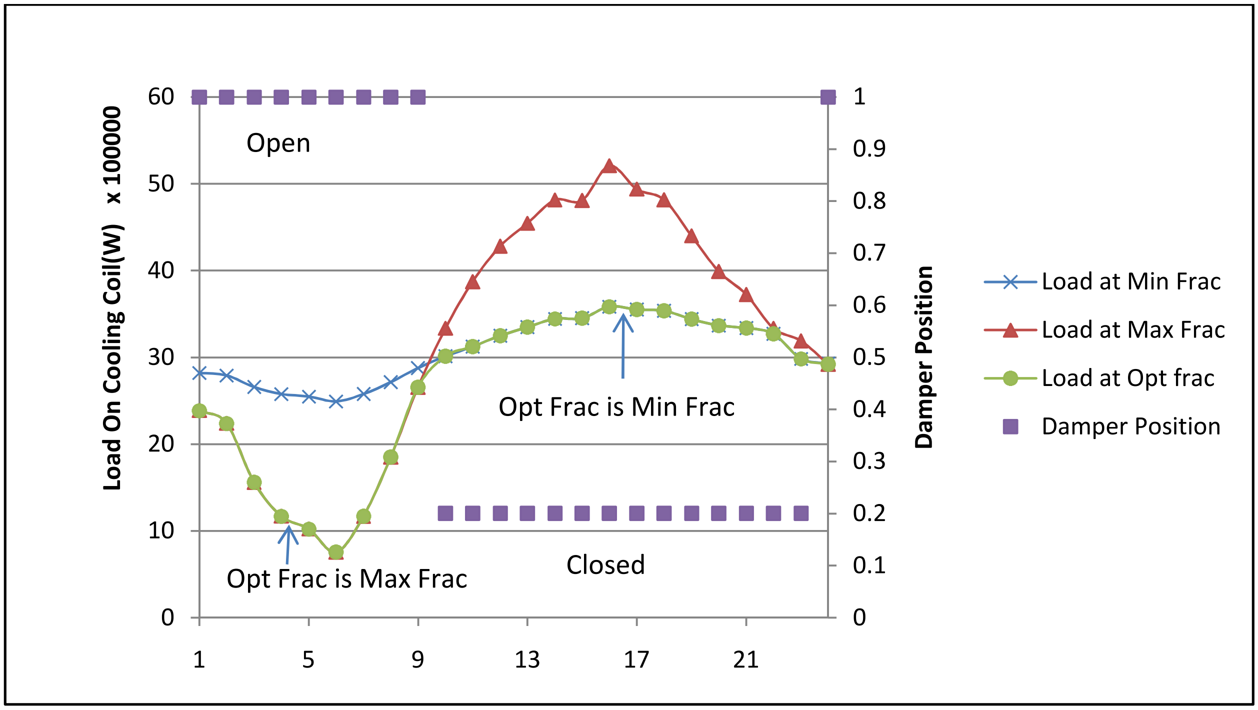


Figure 5.7: Damper positions based on two position detailed coil model based control strategy for April 18

Cooling loads are shown on the primary y-axis while the damper position is shown on secondary y-axis. Figure 5.7 and Figure 5.8 show that the strategy always chooses the fraction of outdoor air that gives the minimum load on the cooling coil.

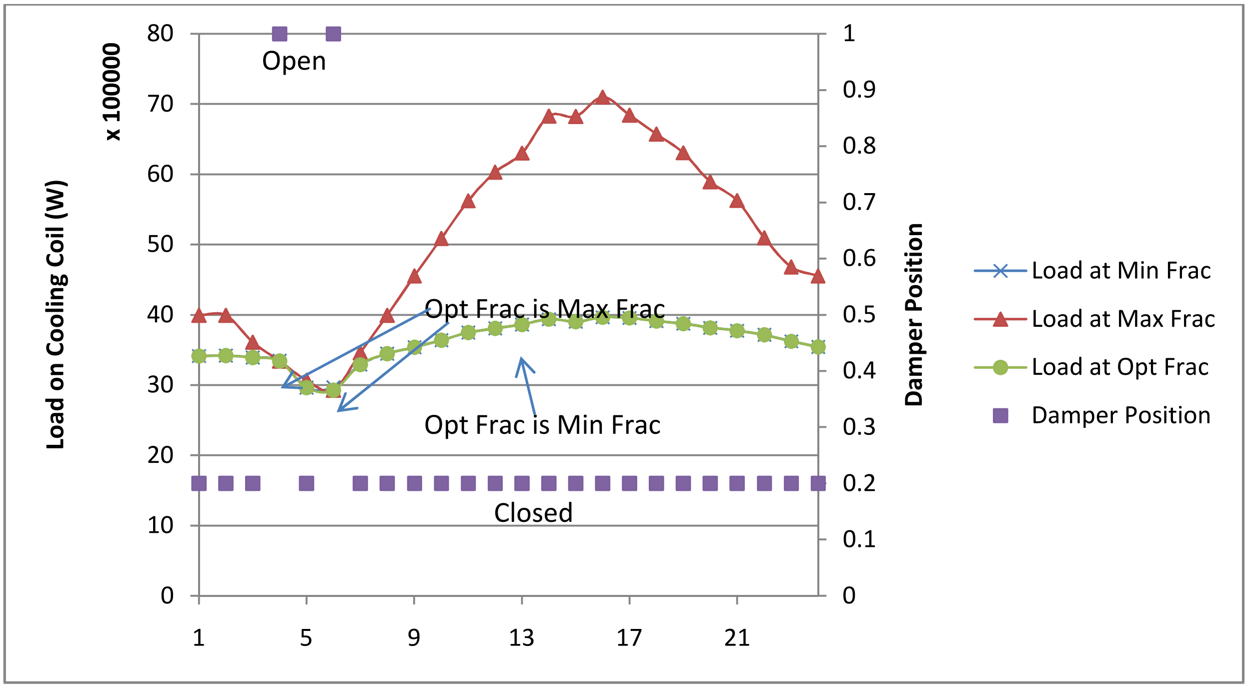


Figure 5.8: Damper positions based on two position detailed coil model based control strategy for June 19

5.1.5 Optimal position detailed coil model based strategy

Optimal position based control strategy works on the same principle as the two position based control strategy. However, the entire damper positions between closed (minimum) and fully open are available for a control decision. Optimal based control uses a one-dimensional optimization routine to calculate the outdoor air fraction that minimizes the load on the cooling coil. The fraction of outdoor air at which the load on the cooling coil is least is chosen as a control decision. Figure 5.9 and Figure 5.10 show the control decision made using the optimal position control strategy.

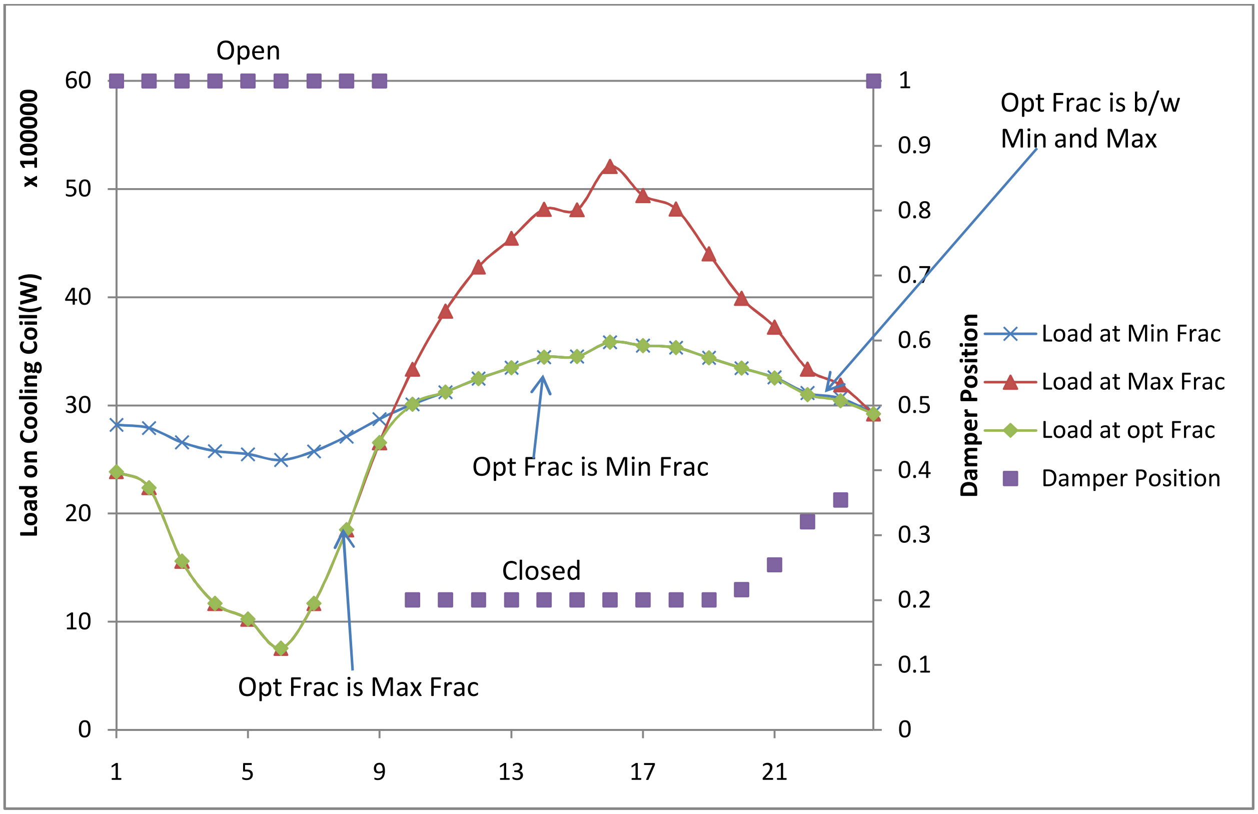


Figure 5.9: Damper position based on optimal position detailed coil model based control strategy for April 18

Table 5.1 shows the values of loads at minimum fraction, maximum fraction and optimum fraction of outside air for 22nd and 23rd hour on 18th April (Figure 5.9) and 4th and 5th hour on 19th June (Figure 5.10).

Table 5.1: Comparison of cooling coil loads for different fractions of outside air

Day and Time	Load at Min frac(kW)	Load at Max frac(kW)	Load at Opt frac(kW)
18th Apr ,22:00	3110.13	3333.7	3098.14
18th Apr ,23:00	3065.3	3188.17	3041.7
19th June , 4:00	3228.2	3339.2	3168.12
19th June , 5:00	3088.5	3069.4	3009.4

The damper position is adjusted for the calculated optimum fraction. From Figure 5.7, its evident that if a two position based control strategy was used the damper would have been closed during the 22nd and 23rd hours on April 18 as load at minimum fraction of outdoor air is less than the load at 100% outdoor air. However, an optimal position based control strategy sets the damper at a position that gives less cooling load than that given by closed damper position.

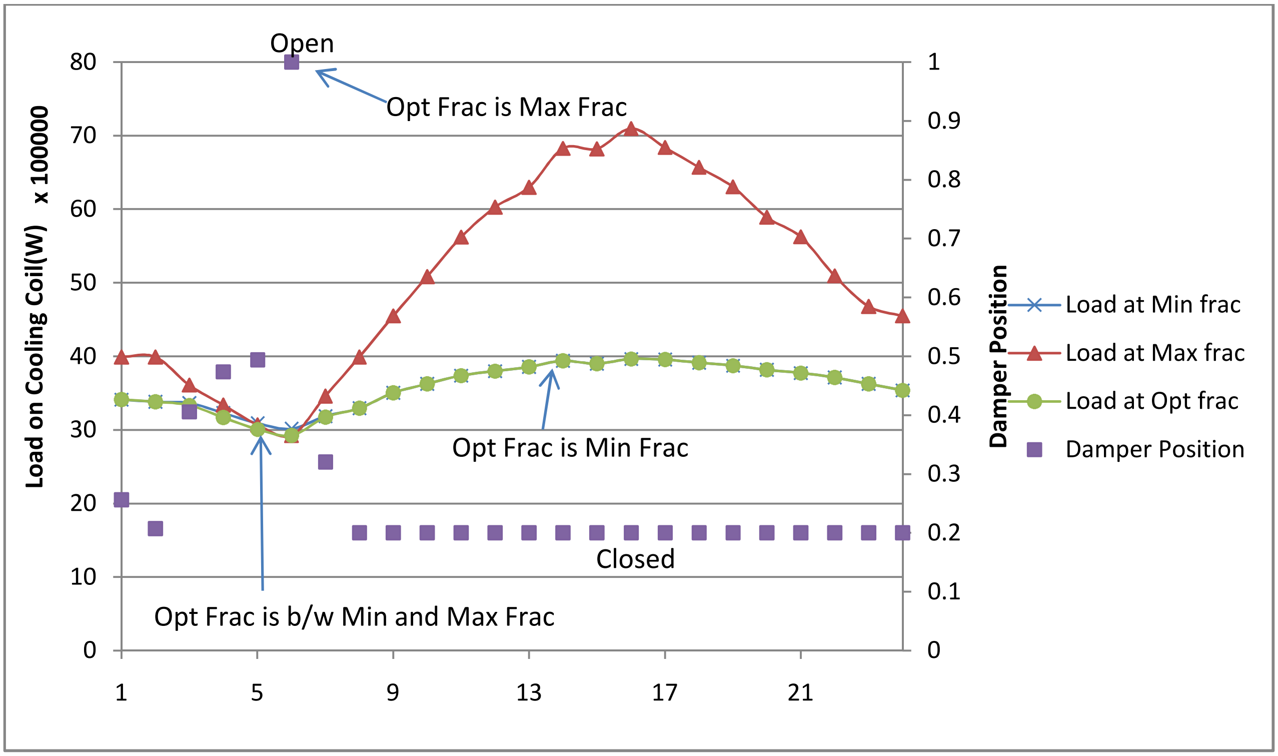


Figure 5.10: Damper position based on optimal position detailed coil model based control strategy for June 19

It is interesting to note that the savings obtained with optimal position based control is observed when outdoor air temperature is slightly warmer than return air temperature. The outdoor air is also drier than return air at these points. Figure 5.11 and Figure 5.12 represent the inferred information from previous statement.

Primary y-axis represents the load on the cooling coils on the hourly basis and secondary y-axis represents the temperature of outdoor and return air.

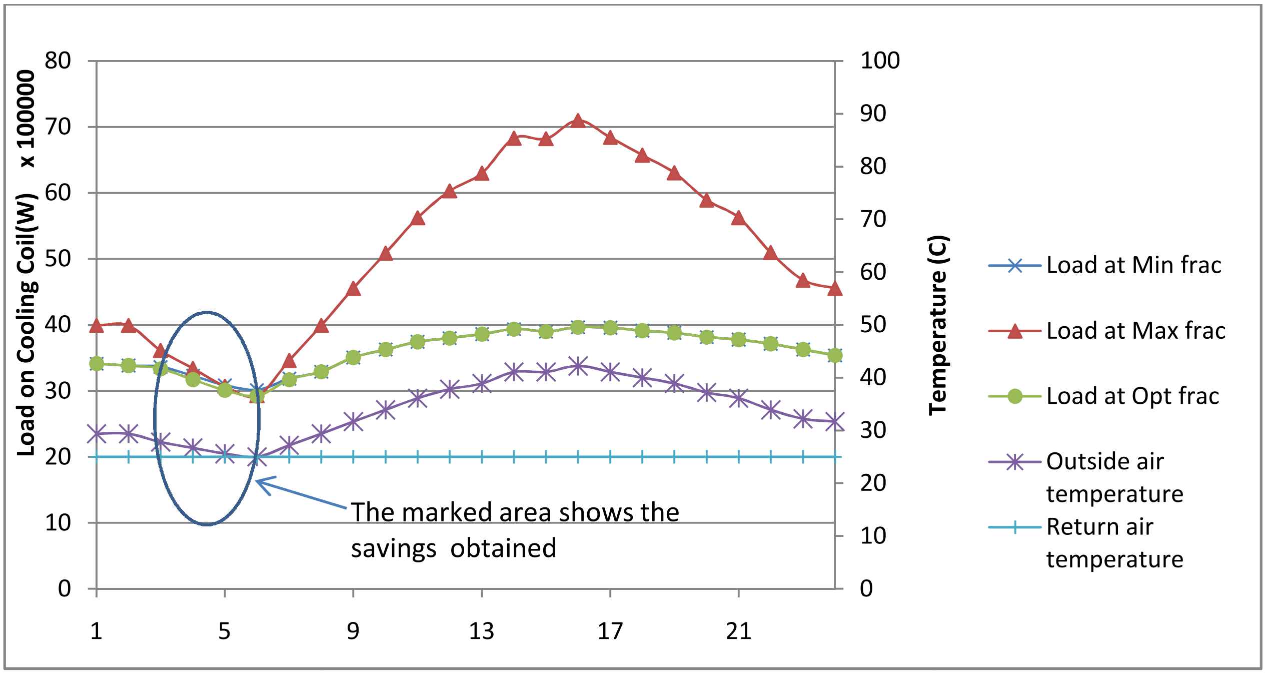


Figure 5.11: Comparison of outdoor and return air conditions when cooling savings are obtained using optimal position model based control strategy for June 19

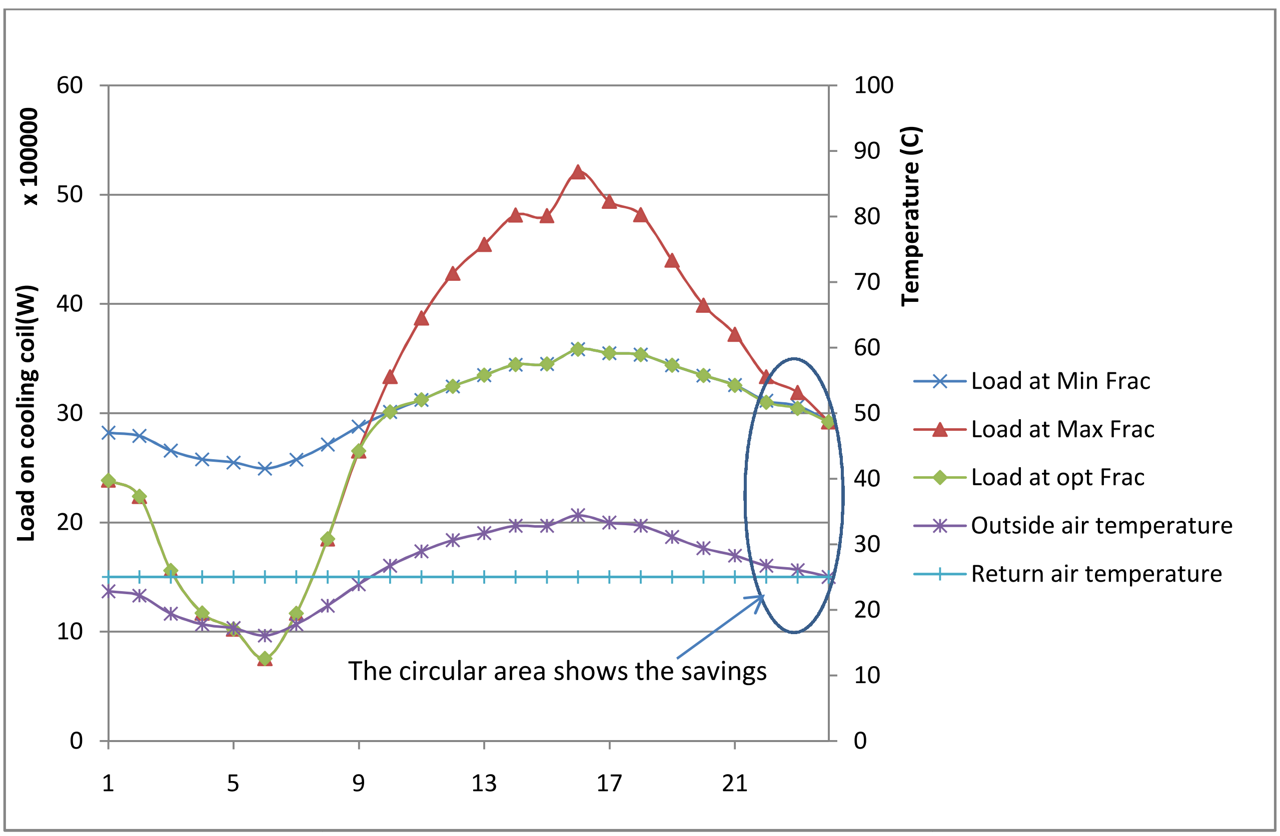


Figure 5.12: Comparison of outdoor and return air conditions when cooling savings are obtained using optimal position model based control strategy for April 18

For verifying that there is indeed savings associated with model based economizer strategies, loads on the cooling coil are compared for conventional and detailed coil controlled model based strategies. The outdoor and return air conditions are represented in Figure 5.13. The return air conditions are approximately the same for every control strategy during this hour. Table 5.2 represents the comparison of loads obtained for 5th hour of 19th June for conventional and model based strategy.

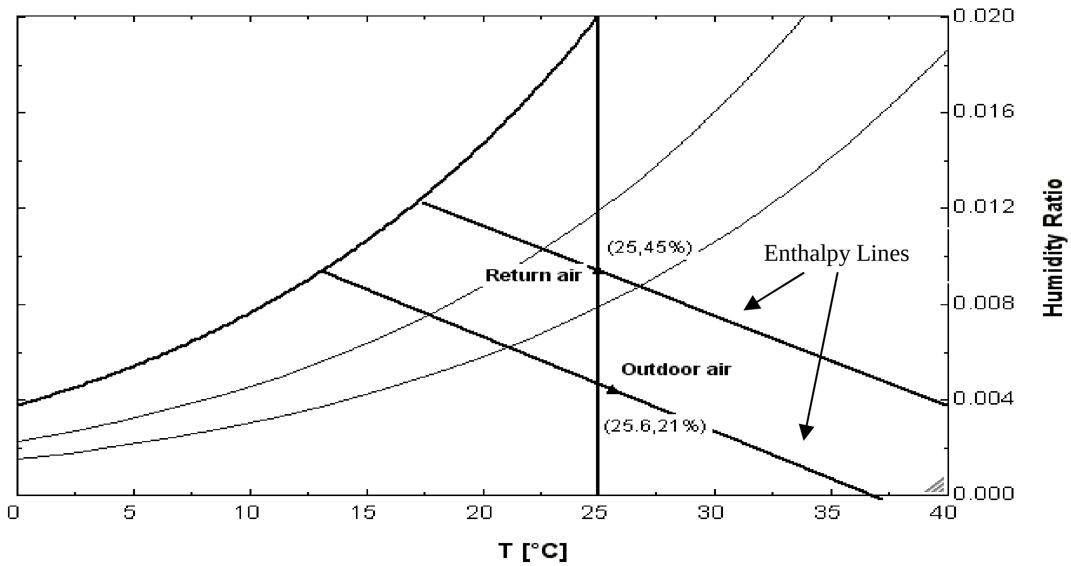


Figure 5.13: Psychrometric chart showing outdoor and return air conditions for 5th hour on 19th June

Table 5.2: Comparison of loads on cooling coil for a single hour on 19th June

Economizer Strategy	Damper Position	Load on cooling coil(KJ/kg)
No Economizer	0.2	13.65
Differential Dry Bulb	0.2	13.65
Differential Enthalpy	1	12.77
Two position detailed coil model	1	12.77
Optimal position detailed coil model	0.494	12.5

5.1.6 Two position bypass coil model based strategy

The strategy also works on the same principle of other model based strategies discussed earlier. However, it uses bypass coil model instead of the detailed coil model to estimate the coil load that is used to make control decisions. The detailed cooling coil model is then used to calculate the actual load on the cooling coil using the control decision made. The simulation is done using the bypass factor of 0.15 for the bypass coil model used in the controller.

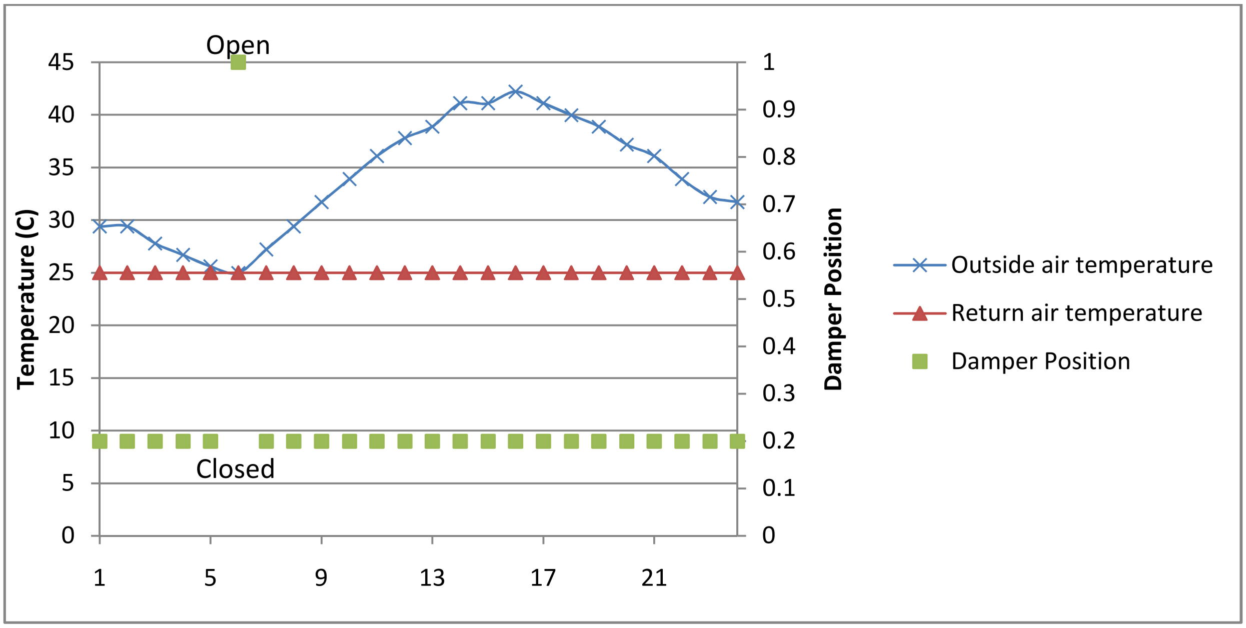


Figure 5.14: Damper position based on two position bypass coil model based control strategy for June 19

Figure 5.14 and Figure 5.15 represent the control decision made by this model based strategy based on minimum load calculation. It can be seen in Figure 5.15 that during hours 1st to 13th, an enthalpy based economizer would keep damper fully open since outdoor air enthalpy is lower than return air enthalpy. However, a model based economizer closes the damper for minimum load on the coil. As pointed out earlier in chapter three, even if outside air enthalpy is lower than return air enthalpy, using 100% outdoor air may increase the sensible load on the coil. If there is no latent load (or a very small latent load), increase in sensible load will increase the total coil load. Figure 5.15 depicts the same thing, 100% outdoor air increases the sensible load on the coil while a minimum fraction of outdoor gives a lower load on the coil.

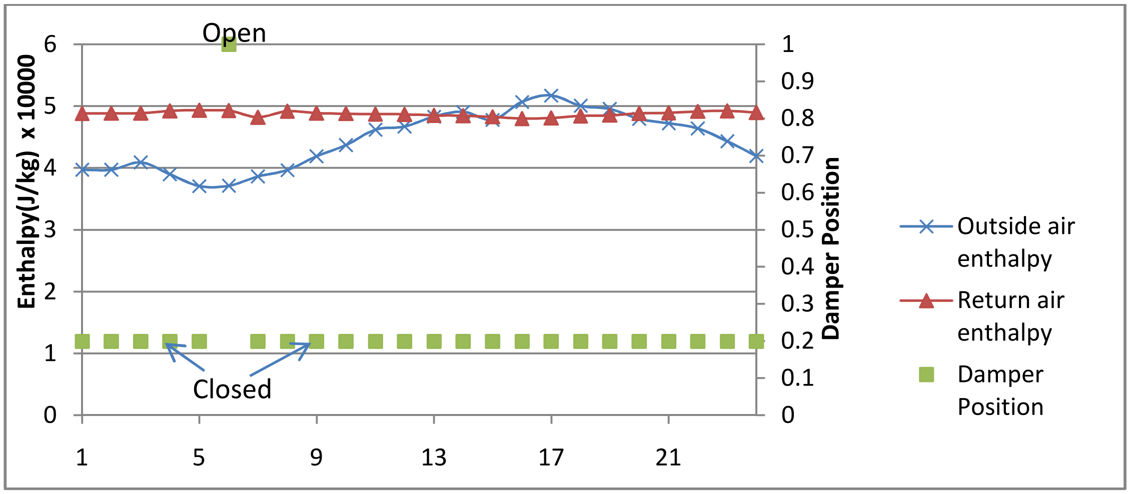


Figure 5.15: Damper position based on two position bypass coil model based control strategy for June 19

5.1.7 Optimal position bypass coil model based strategy

This modeling strategy uses the simplified bypass coil model in the controller to determine the optimum outdoor air fraction to minimize the load on the cooling coil. Figure 5.16 shows the damper position for variation in temperature on a day.

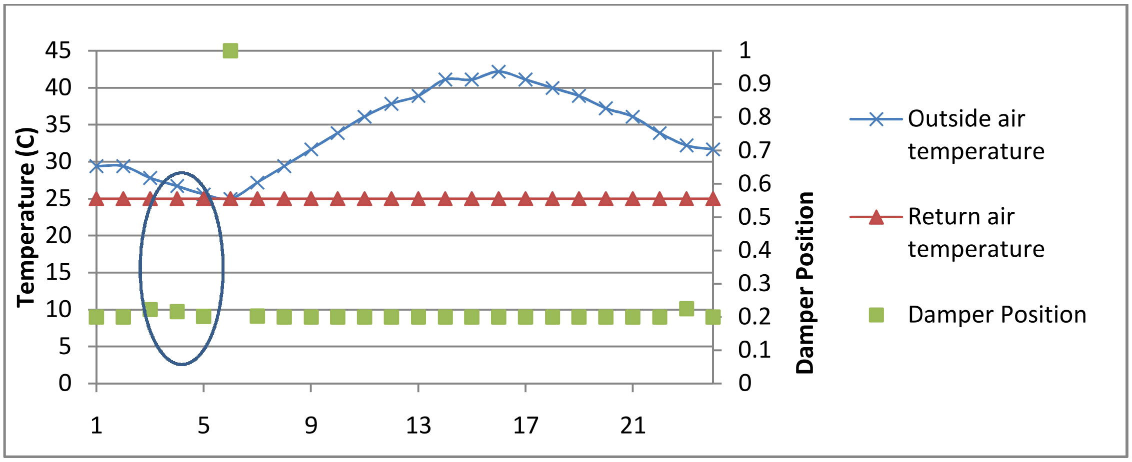


Figure 5.16: Damper position for optimal position bypass coil model based control strategy for June 19

As stated earlier the optimal position model based strategy as found to have least cooling loads at outside air being warmer than return air. It can be seen in Figure 5.16 that the damper position is different from two extreme positions during hours 3rd to 5th .

5.2 Verification of simulation results

For verification purposes, the results obtained from EnergyPlus are compared with Seem and House (2009) simulation results. Annual simulation is done on seven different cities chosen to obtain cumulative cooling coil energy for an annual run and annual peak cooling coil load. The output from EnergyPlus is in Joules, thus it is converted to KJ/kg of dry air for direct comparison with results reported by Seem and House (2009).

Table 5.3 shows the annual cooling coil energy consumption for 7 USA locations obtained from EnergyPlus. As expected, optimal position model based strategies perform best for all cities except Houston. For Houston, two position model based strategies performs slightly better than optimal position model based strategies.

Table 5.3: Cumulative cooling coil energy for seven US locations using seven different economizer strategies

EnergyPlus Results	Cumulative Annual Cooling Coil Energy (KJ/kg of dry air)						
Economizer Strategy	Phoenix	Chicago	Charlotte	Houston	Los Angeles	Newyork	Albuquerque
No Economizer	122947.3	101031.8	119783.4	140427.1	120926.4	108410.9	98597.6
Differential Dry Bulb	87997.4	47306.2	78353.5	121920.9	57111.2	57691.4	41006.1
Differential Enthalpy	88786.2	44199.1	70285.5	107019.2	56548.2	51292.4	41170.7
Two position bypass model based	87630.4	44152.7	70238.7	**107034.9**	56468.6	51294.5	40992.4
Optimal position bypass model based	87598.3	44151.2	70239.0	**107038.2**	56468.2	51294.7	40992.2
Two position detailed model based	87527.2	44133.8	70220.6	**106997.5**	56456.4	51264.0	40702.9
Optimal position detailed model based	87402.9	44131.2	70209.6	**106999.6**	56453.3	51262.1	40667.3
Air Mass flow rate(kg/s)	240.4	215.6	215.7	203.1	206.5	216.3	190.1

Two factors may contribute to this scenario. First, the optimal control strategy optimizes the control decision only for the current time step. Optimization over a longer time period could change the results. Second, the model based strategies result in different return air conditions which in turn result in different cooling coil loads. To verify that this is only due to the difference in return air humidity ratio, humidity and enthalpy, a model test is performed with the same outdoor and return air conditions for Houston. It was found that the optimal position model based control saved almost 4KJ/kg over an annual period simulation than two position model based control. Thus, under all identical return air conditions, optimal position model control would show higher energy savings than any other economizer strategies.

5.2.1 Comparison of simulation results with Seem and House (2009)

Table 5.4 shows Seem and House (2009) simulation results for cooling coil energy. These results are obtained using MATLAB and bypass cooling coil model explained earlier. The return air temperature and supply air temperature is fixed and the humidity ratio of return air is determined by addition of constant (0.0015) to the supply air humidity ratio. The bypass factor used for simulation by Seem and House is 0.15.

Table 5.4: Seem and House (2009) simulation results for cooling coil energy with five economizer strategies for seven US locations

MATLAB Results	Cumulative Annual Cooling Coil Energy (KJ/kg of dry air)						
Economizer Strategy	Phoenix	Chicago	Charlotte	Houston	Los Angeles	New York	Albuquerque
No Economizer	121227	99119	117827	139472	120,327	106,953	96149
Differential Dry Bulb	86994	46236	76775	121147	55,319	57,308	40166
Differential Enthalpy	88283	43329	69019	106328	54,822	50,925	40527
Two position bypass model based	86537	43257	68957	106300	54,691	50,886	39847
Optimal position bypass model based	86141	43228	68920	106294	54,688	50,864	39642

Percentage error between the values obtained from EnergyPlus and Seem and House (2009) simulations are calculated. The percentage error is the deviation based on Seem and House results.

% error = (EnergyPlus results – Seem and House results)/ Seem and House Results

Table 5.5 shows the percentage deviation for cooling coil energy.

Table 5.5: Percentage error between EnergyPlus and Seem and House (2009) simulation results for cooling coil energy

	Error in Cumulative cooling coil energy(%)						
Economizer Strategy	Phoenix	Chicago	Charlotte	Houston	Los Angeles	New York	Albuquerque
No Economizer	1.42	1.93	1.66	0.68	0.50	1.36	2.55
Differential Dry Bulb	1.15	2.31	2.06	0.64	3.24	0.67	2.09
Differential Enthalpy	0.57	2.01	1.83	0.65	3.15	0.72	1.59
Two position bypass model based	1.26	2.07	1.86	0.69	3.25	0.80	2.87
Optimal position bypass model based	1.69	2.15	1.92	0.70	3.26	0.85	3.41

The percentage error ranges from is 0.5% to 3.41%.

Similarly, the annual peak cooling coil loads are also compared. Table 5.6 shows the peak cooling coil loads obtained for seven cities with seven economizer strategies.

Table 5.6: EnergyPlus results for annual peak cooling coil loads

EnergyPlus Results	Peak Load (KJ/kg)						
Economizer strategy	Phoenix	Chicago	Charlotte	Houston	Los Angeles	New York	Albuquerque
No economizer	21.34	22.84	22.42	22.90	18.20	21.90	17.42
Differential dry bulb	34.28	35.49	37.18	38.42	25.51	36.78	19.09
Differential enthalpy	28.37	22.84	22.42	22.90	18.51	21.90	24.48
Two position bypass model based	21.34	22.84	22.42	22.90	18.98	21.90	17.42
Optimal position bypass model based	21.34	22.84	22.42	22.90	18.87	21.90	17.42
Two position detailed model based	21.34	22.84	22.42	22.90	18.98	21.90	17.42
Optimal position detailed model based	21.34	22.84	22.42	22.90	18.98	21.90	17.42

Excluding the no economizer case, model based strategies give the lowest annual peak loads of all the other economizer strategies. Temperature based control gives the highest annual peak loads except for Albuquerque. Seem and House results for annual peak loads are shown in Table 5.7.

Table 5.7: Seem and House results for annual peak cooling coil loads

MATLAB Results	Peak Load (KJ/kg)						
Economizer strategy	Phoenix	Chicago	Charlotte	Houston	Los Angeles	New York	Albuquerque
No economizer	21.23	22.54	22.14	22.81	18.21	21.9	17.47
Differential dry bulb	33.66	35.15	37.01	39.08	25.31	37.01	19.45
Differential enthalpy	29.01	22.54	22.14	22.81	18.7	21.9	25.1
Two position bypass model based	21.23	22.54	22.14	22.81	19.19	21.9	17.92
Optimal position bypass model based	21.23	22.54	22.14	22.81	19.19	21.9	17.92

The percentage error deviation in annual peak loads predicted using EnergyPlus and Seem and House results range from -2.8% to 1.8%. Figure 5.17 and Figure 5.18 shows the comparison for cumulative cooling coil energy and annual peak cooling loads respectively for EnergyPlus and Seem and House (2009) results. The EnergyPlus results are presented on y-axis and the Seem and House (2009) results are shown on x-axis.

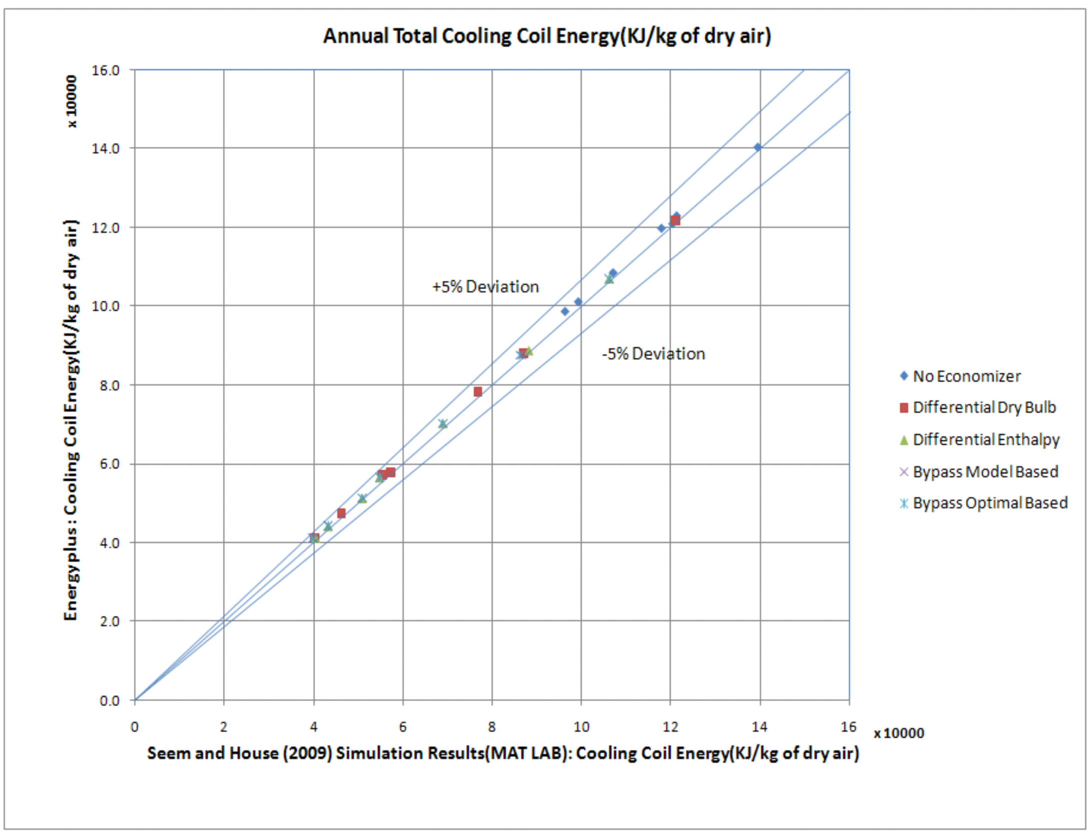


Figure 5.17: Comparison of annual cooling coil energy

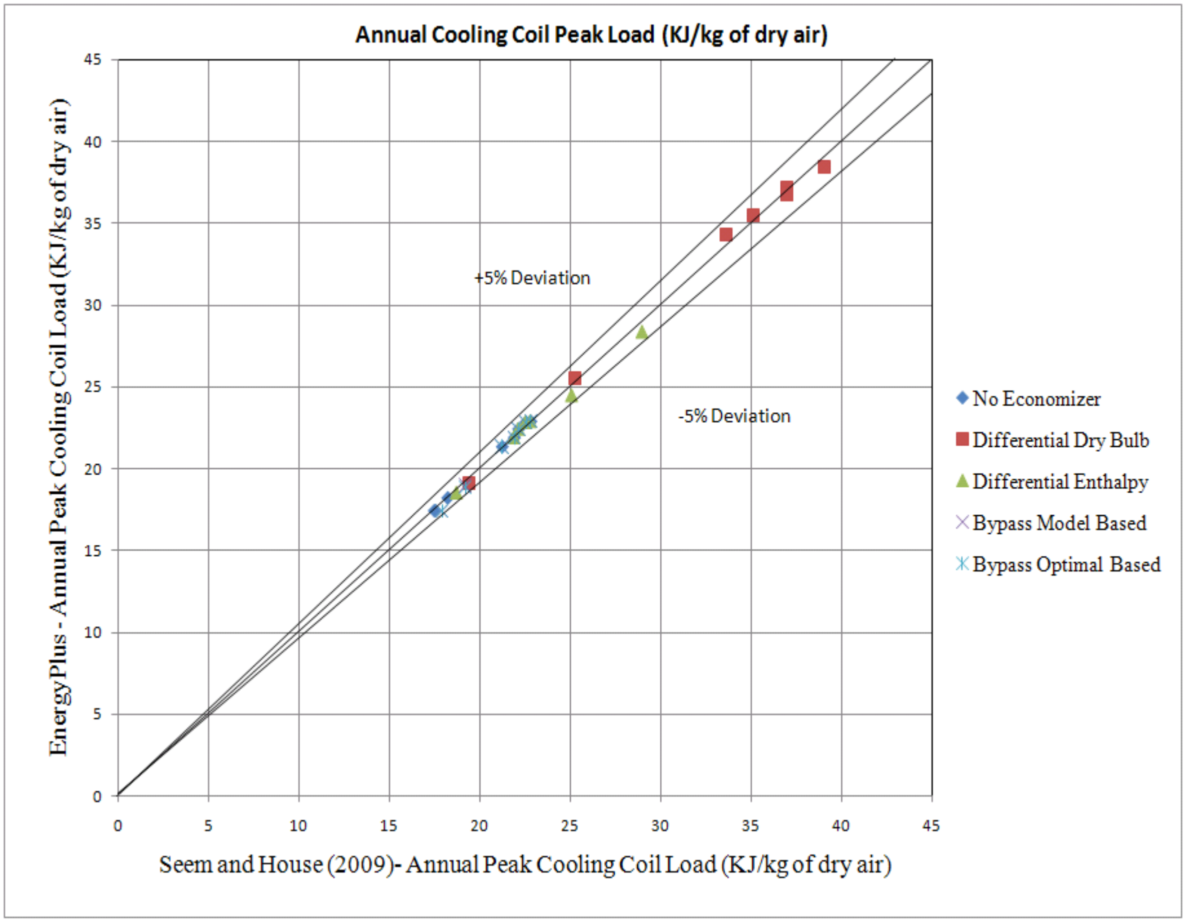


Figure 5.18: Comparison for annual peak cooling coil load

5.2.2 Comparison of coil models

The bypass coil model results were obtained for bypass factor of 0.15. For the best estimation of bypass factor that could be used to replicate the results obtained by detailed coil model, an EES model was developed. The EES model uses the cooling loads and coil conditions obtained from detailed coil model in EnergyPlus to estimate the load using the bypass coil model. The model minimizes the squared sum of difference between the estimated bypass coil load and actual detailed coil load by fitting a bypass factor. A data set of 30 different conditions was picked from outputs obtained from detailed coil model simulation based on wet, partially wet and dry coil. A bypass factor of 0.016 was fitted to according to the EES model. When the annual simulation were performed again with the fitted bypass factor for bypass coil model based economizer strategies, it was observed that the total cooling coil energy predicted by a bypass coil controller in EnergyPlus is within 1% of that predicted by a detailed coil controller. Moreover, the annual peak loads calculated by bypass model and detailed model based strategies are almost always the same. Thus, if using a model based economizer strategy for damper control for implementation, the detailed coil controlled model could be easily replaced by bypass coil controlled model. The bypass coil model is easier to implement and would provide approximately the same savings that could be realized by using detailed coil model based controls.

To investigate the modeling error due to the change in bypass factor simulations were also performed using bypass factors ranging from 0.05 to 0.25 in increments of 0.05. Although it was observed that as the bypass factor increased, the number of hours in a year for 100% outside air flow also increased as shown in Figure 5.19, the effect on annual cooling coil energy was small. It was observed that the annual cooling coil energy predicted by using these bypass factors are within 1% of each other as shown in Table 5.8.

Table 5.8: Comparison of cooling coil energy on changing bypass factor for Chicago

Bypass factor	Total cooling coil energy (MWh)
0.05	2568
0.1	2569
0.15	2570
0.2	2571
0.25	2573

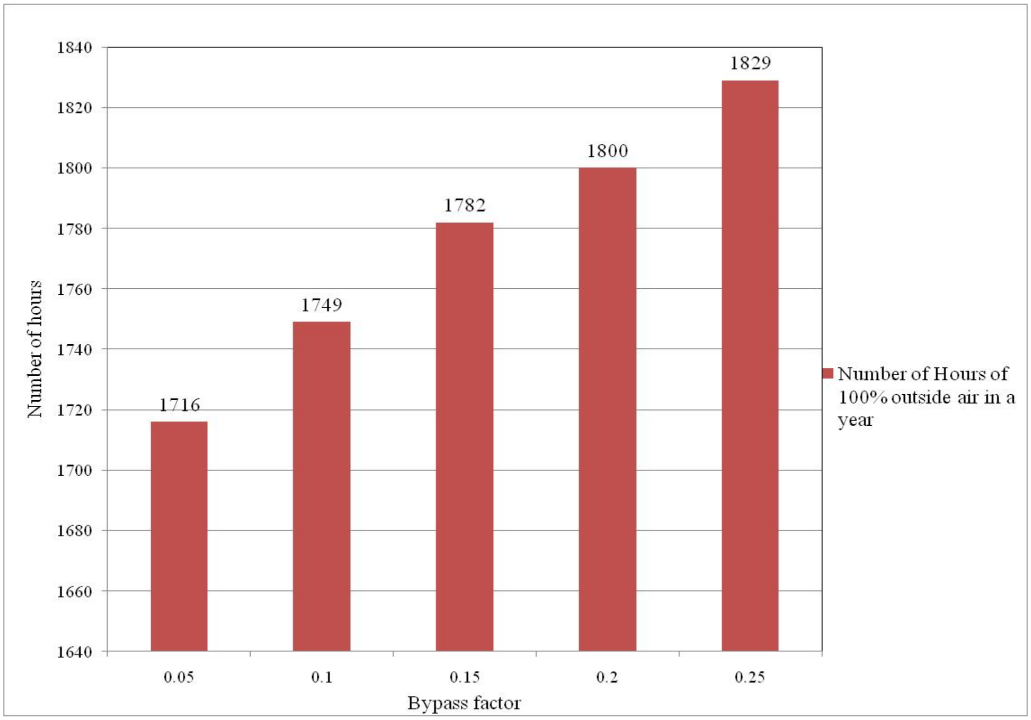


Figure 5.19: Number of hours in a year for fully open damper for Chicago

As the coil bypass factor increases, the leaving air dry bulb temperature of the coil decreases and the leaving air humidity ratio of the wet coil decreases as well. For a dry coil, the leaving air dry bulb temperature decreases as the bypass factor increases while the humidity ratio remains constant. When the bypass factor in the control model differs from the system cooling coil bypass factor, the control point calculated by the bypass factor model can fall into one control region of the psychrometric chart; and the actual system operating point can fall into another region. This effect is shown in Figure 5.20. Point P represents the return air conditions. The 'actual point' represents the cooling coil leaving air conditions, and the 'control point' represents the leaving air conditions calculated by bypass coil model in the controller. (Refer Figure 3.1 for explanation of control regions on psychrometric chart)

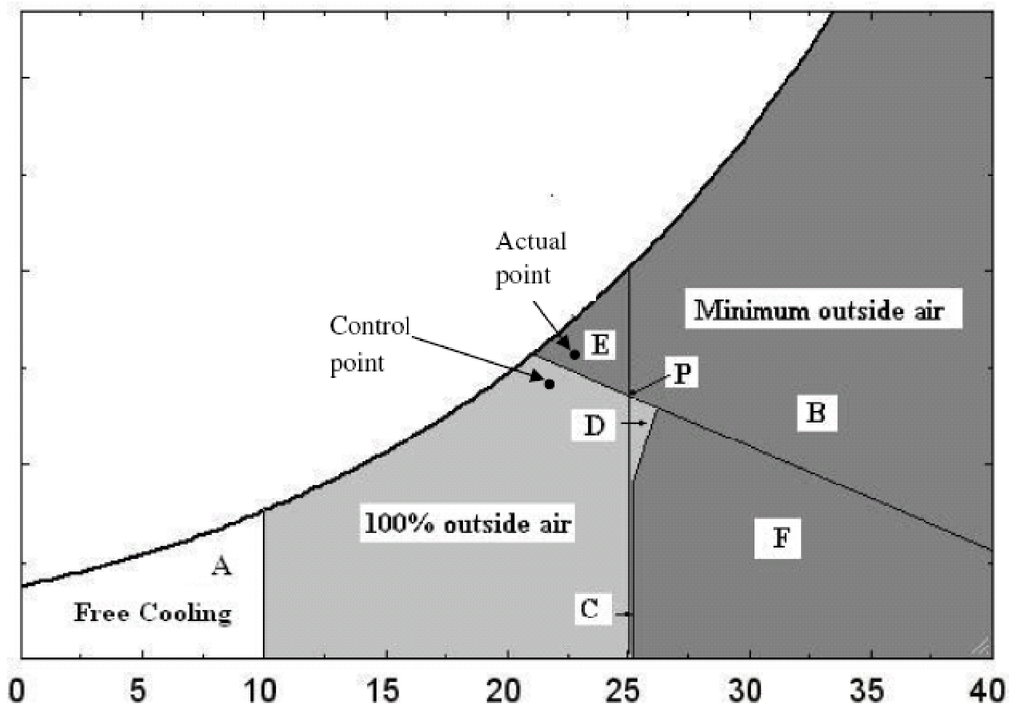


Figure 5.20: Effect of bypass factor on control decision

The key point is that for controller model bypass factors greater than system coil bypass factor, the effect will always be to open the damper when it should be closed. One of the examples of that sort is given in Table 5.9. For controller model bypass factors less than the system coil bypass factor, the effect will always be to close the damper when it should be open. Therefore, the bypass factor should be conservatively estimated, but in any case the impact on annual energy is small as long as the estimated bypass factor is reasonable.

Table 5.9: Difference in control decision due to change in bypass factor

	Bypass factor = 0.15	Bypass factor = 0.25
OA Temp (^{o}C)	18.55	18.55
OA Relative humidity (%)	87.5	87.5
RA Temp (^{o}C)	22.97	22.97
RA Relative humidity (%)	50.8	50.8
Damper control position	0.2	1.0
Cooling load on coil (kW)	1237	1387

5.2.3 Analysis of results

The percentage deviation in estimated cooling coil energy and annual peak loads between EnergyPlus and Seem and House (2009) results are already mentioned earlier. Model based strategies gave the least annual peak loads and have a potential to save cooling energy. The differences in estimated load and cooling energy values for Seem and House results and EnergyPlus results may be due to following reasons:

1. Seem and House (2009) simulation results are obtained using the bypass cooling coil model based on contact mixture analogy for load and energy calculation over the coil. The coil is either considered dry or fully wet for bypass cooling coil model. EnergyPlus uses a detailed coil model to predict the total cooling load on the coil and thus the total cooling coil energy. This leads to difference in the values calculated for load and energy by Seem and House (2009) simulation and EnergyPlus simulation results.
2. The zone time step in EnergyPlus and Seem and House (2009) simulation is one hour. However, time steps could be reduced in EnergyPlus to system time steps for convergence (as less as 1 minute). The load on the cooling coil obtained is multiplied with the time step period to obtain the cooling coil energy in EnergyPlus. Since the output is hourly, the cooling coil energy is summed up for the hour in joules. The comparison of energy is based on KJ/kg , thus the cooling coil energy obtained in Joules from EnergyPlus is divided by the mass flow rate of air, an hour time step in seconds(3600) and 1000 to convert it in KJ/kg of dry air. This may not be representative of the whole year simulation and thus may cause difference.

3. As pointed out earlier, the annual cooling coil energy in joules is divided by mass flow rate of air for unit conversion. The air mass flow rate used is the averaged mass flow rate of year. It does not represent the air flow rate for the whole year. The mass flow rate of air changes with density which depends on temperature. There may not be a large difference in the flow rates but considerable enough to explain the difference.
4. The Seem and House (2009) simulation results are obtained at atmospheric pressure all year round. EnergyPlus however, uses the pressure of outdoor air obtained from the weather files. This causes a noticeable but not significant difference in calculating the enthalpies of outdoor air and return air. The difference becomes more relevant when outdoor and return air enthalpies are close to each other.

However, stating that optimal position based control strategy would give best results in most of the cases and thus should be used with economizers would be a premature statement. It is important to realize that in realistic building management systems, the sensors measuring the temperature and relative humidity are associated with sensor errors. It thus becomes important to see the effects of sensor errors on control decisions made by various economizer strategies. The following section is dedicated to study modeling strategies again with imperfect sensors.

5.3 Verification with Sensor errors

For this investigation it was necessary to verify the modeling strategies with sensor errors. Temperature and relative humidity sensor errors were considered. Twenty eight different types of error combinations were chosen on outside and return air conditions for modeling. The errors were selected to replicate the results represented by Seem and House (2009). The range of dry bulb error was -2 ºC to +2 ºC; the range of relative humidity error was -10% to +10%. Three cities were modeled with five economizer strategies with these sensor errors. The system is same as shown in Figure 5.1. The results obtained through simulation in EnergyPlus are compared with the results obtained from Seem and House (2009). It was found that EnergyPlus results for cumulative cooling coil energy and peak cooling coil loads agree to within ±5% except for 2 results out of 840 total results obtained. The annual peak cooling coil loads were found to differ by more than 5% for two cases in phoenix with differential enthalpy control. It was observed that the models produced different control decisions when outdoor air enthalpy was very close to return air enthalpy. Although the models predicted nearly the same return air conditions, they fell into different control region on the psychrometric chart (Figure 1.2).

The following chapter now discusses and describes the buildings chosen for parametric study. The simulation is done with floating return air conditions and HVAC system is equipped with variable air volume systems.

CHAPTER 6

6. Parametric study

Model evaluation and verification of EnergyPlus simulation results of the conventional and the model based economizer strategies have given confidence to perform a detailed study with these economizer strategies. This chapter focuses on analyzing the annual cooling coil energy usage and annual peak loads obtained for different economizer strategies across fifteen different cities in US. The fifteen cities represent different climatic zones in USA. Table 6.1 shows different cities with their respective climatic zones.

Table 6.1: Cities representing different climatic zones

Climate Zone	Description	City
1A	Very Hot - Humid	Miami, FL
2A	Hot - Humid	Houston, TX
2B	Hot - Dry	Phoenix, AZ
3A	Warm - Humid	Charlotte, NC
3B	Warm - Dry	Los Angeles, CA
3C	Warm - Marine	San Francisco, CA
4A	Mixed - Humid	New York, NY
4B	Mixed - Dry	Albuquerque, NM
4C	Mixed - Marine	Seattle, WA
5A	Cool - Humid	Chicago, IL
5B	Cool - Dry	Denver, CO
6A	Cold - Humid	Minneapolis, MN
6B	Cold - Dry	Cheyenne, WY
7A	Very Cold - Humid	Phillips, WI
7B	Very Cold - Dry	Jackson, WY

Three different buildings are chosen for the detailed study. The buildings represent a large office, large hotel and large hospital. Annual EnergyPlus simulations are performed using conventional and detailed coil model based economizer strategies for controlling outside air damper position.

For the purpose of parametric testing, EnergyPlus input files for the three buildings are obtained from DOE. These input files contains the geometry, construction materials, HVAC equipment, non HVAC equipment and various schedules (lighting, occupancy etc) associated with the buildings. The location of building and the weather file associated with the location are provided separately. All the economizer strategies are simulated for an annual run through all the buildings for all the locations. The simulation is set to decide the return air conditions unlike the verification runs where return air conditions are fixed to a constant temperature and separately calculated humidity ratio. It is understood that return air conditions would vary based on control decisions made by different economizer strategies. However, that makes the comparison more realistic in a sense considering that one of those strategies would be working independently over the building system throughout the year. The outputs are obtained hourly. Annual cumulative cooling coil energy consumption and annual peak load are the focused variables. Results are also obtained with sensor errors on temperature and humidity measuring sensors.

6.1 Buildings

The basic building models with system components are described in coming sections. The buildings are classified based on their cooling needs.

6.1.1 Office building

The office is a 12 story building with basement. The area of the building is 42757 m^2. Each story is a divided into five zones with four perimeter zones and one core zone. There is only one outside air controller for the building. The core zone of the building is comparatively larger than the perimeter zones. Furthermore, the building is divided in four sections named as basement, bottom, middle and top. The bottom is the first floor, the middle contains ten floors and the top is the twelfth floor. The air primary loop is a variable air volume system with reheaters. The heating coils are connected to gas fired boilers which provide hot water for heating the air. The occupancy schedule determines the number of people in the building during the day. Constant values for internal heat gains as lights, electric loads are added depending on their schedule of operation.

The total zone load profile for office buildings is shown in Figure 6.1. This is obtained after an annual simulation of the office building in Chicago.

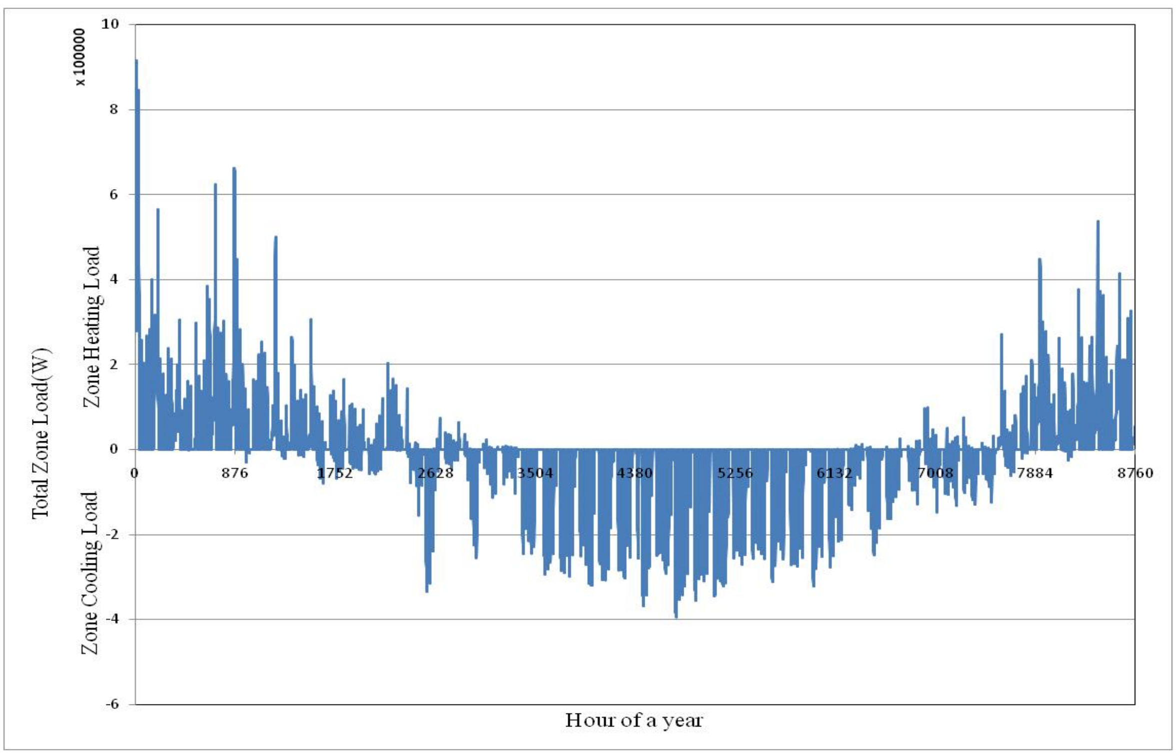


Figure 6.1: Load profile for an office building in Chicago

It can be seen that the building is heating dominated in early and late part of the year and cooling dominated in the middle. Since this is an office building, the loads are zero on holidays when there is no occupancy.

6.1.2 Hotel building

The hotel is a 6 story building with 179 rooms and a laundry facility. The area of the hotel is 9366 m^2. The air loop contains variable air volume system with reheaters at different zones. The zones include lobby, rooms, kitchen, laundry, balcony, dining areas and conference room. Air heating is done using natural gas fired boilers. Different internal gains and occupancy schedules are given for every zone.

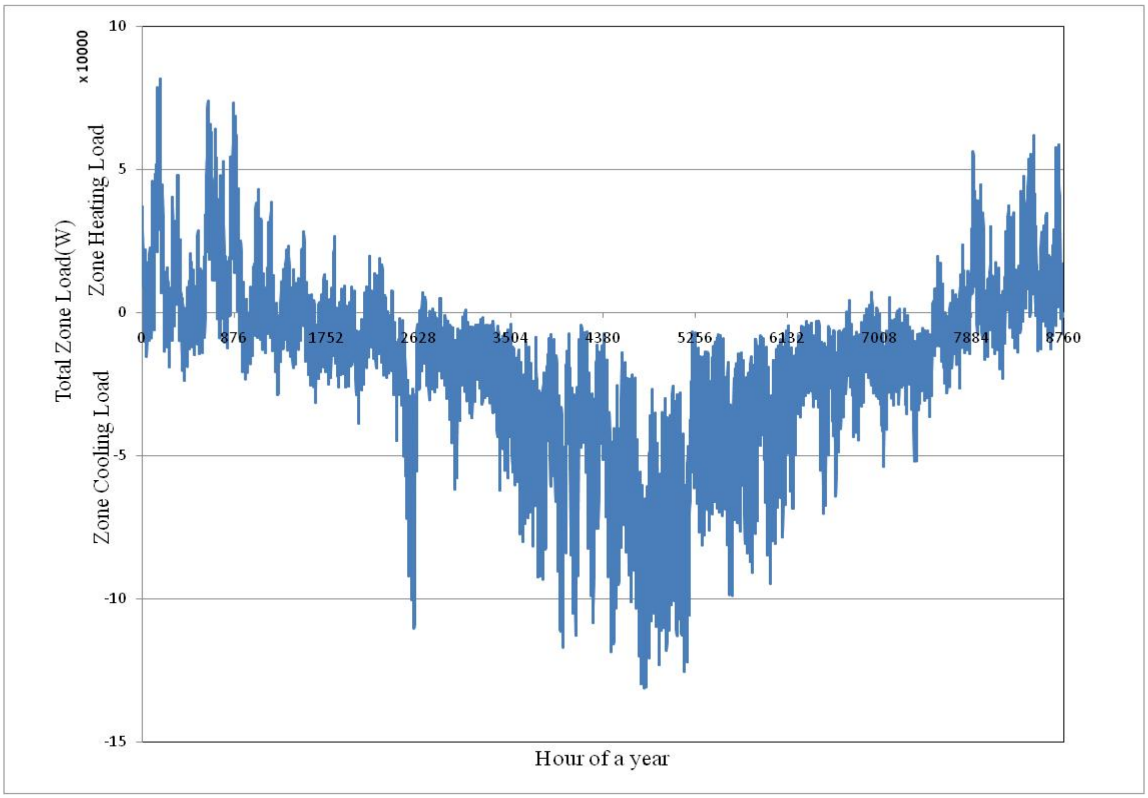


Figure 6.2: Load profile for a hotel building in Chicago

The load profile for a hotel building is shown in Figure 6.2. The building is cooling dominant and has very less hours in a year when heating is required. The kitchen requires year round cooling due to internal heat gains throughout the year.

6.1.3 Hospital building

The hospital is a 5 story building with a basement. The area of the building is 18697 m^2. The zones are classified as basement, emergency rooms, operation rooms, intensive care units, kitchen, dining areas, labs, patient rooms, lobby, office, corridors and nurse rooms. There are two primary air loops in the building. The primary system is variable air volume system with reheaters. The internal gains include lights, electric loads and gas loads. The boilers are gas fired and the cooling coils get chilled water from a purchased source which means that cooling coils have capacity to provide enough cooling to cool entering air conditions to coil to set point temperature.

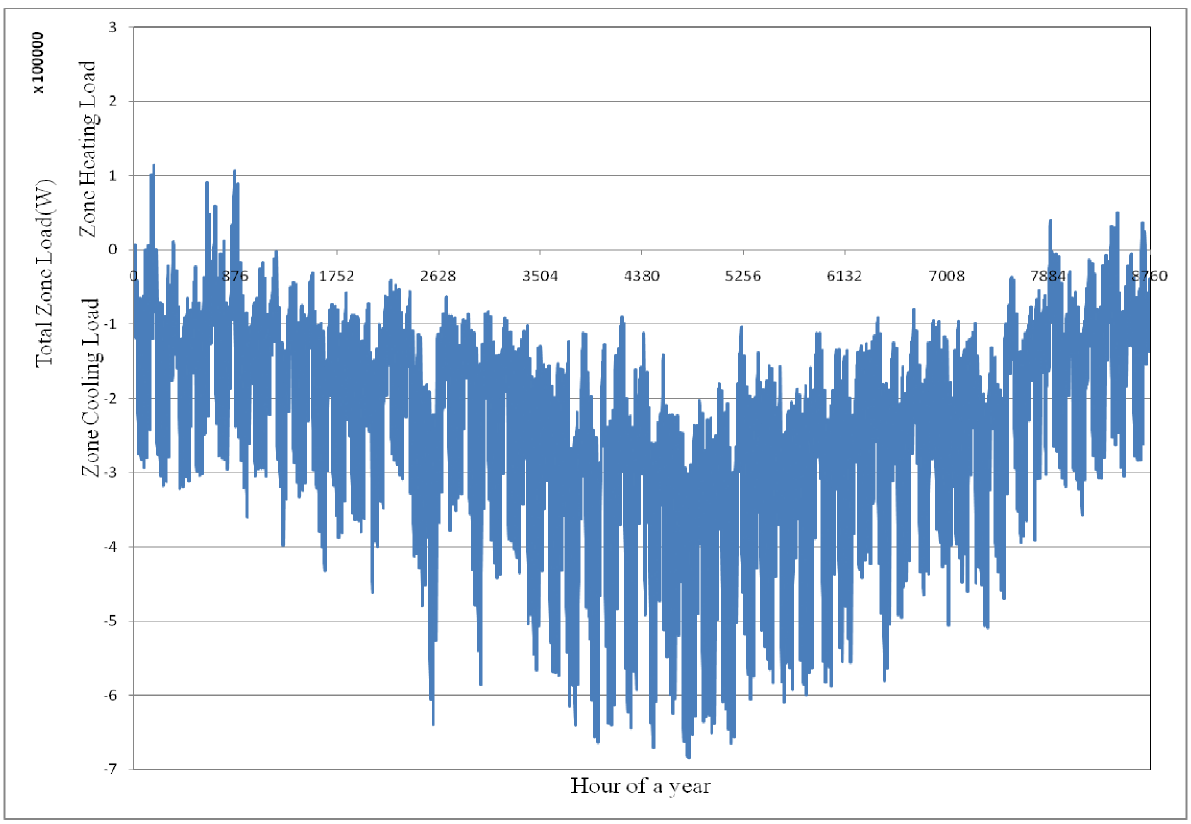


Figure 6.3: Load profile for a hospital building in Chicago

In Chicago, the hospital building is entirely cooling dominated as shown in Figure 6.3. The kitchen, labs and the rooms with medical equipment that produce a lot of heat require year round cooling.

The coming chapters discuss the impact on energy savings and design by using economizer strategies in these buildings.

CHAPTER 7

7. Impact of Economizer Strategies on Annual Cooling Coil Energy

The building types described in chapter 6 are simulated with conventional and detailed model based economizer strategies for perfect and imperfect sensors. The discussions in this chapter are based on the savings in cooling coil energy that can be achieved using the economizer strategies.

7.1 Perfect sensors

Cumulative cooling coil energy over an annual run is plotted for fifteen cities for different economizer strategies.

7.1.1 Office building

The results for large office building are shown in Figure 7.1, Figure 7.2, and Figure 7.3.

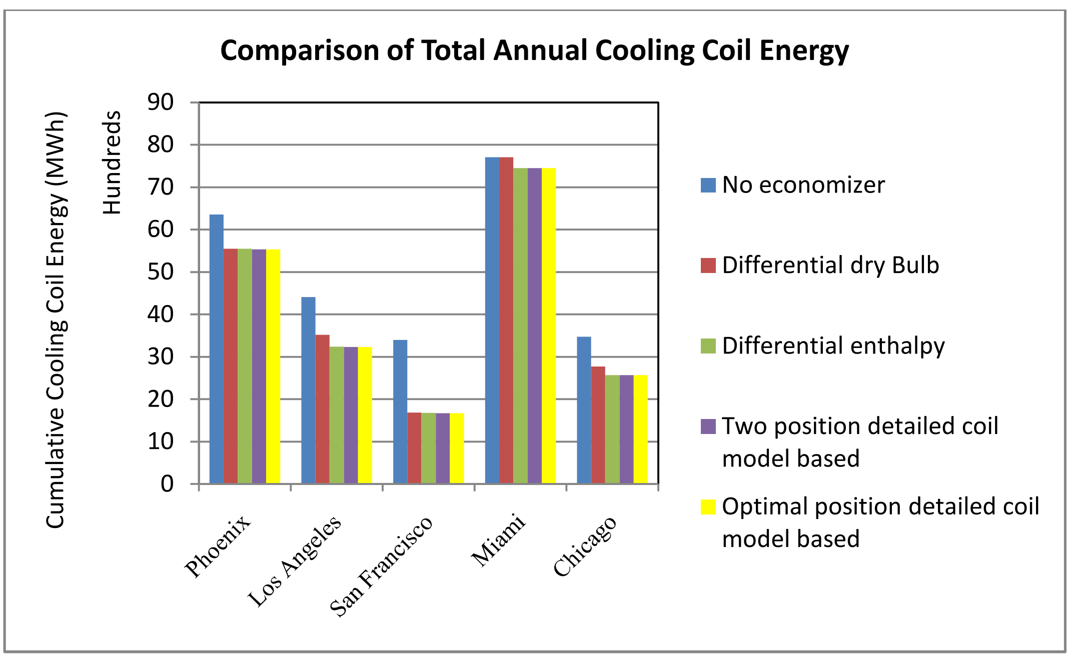


Figure 7.1: Comparison of annual cooling coil energy - I

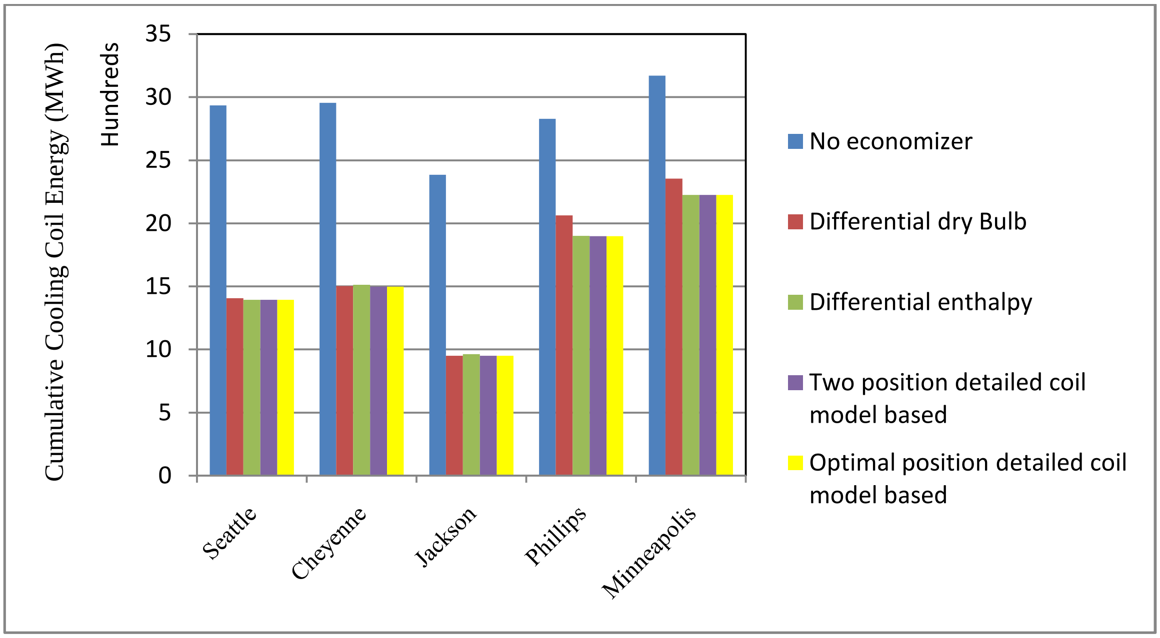


Figure 7.2: Comparison of annual cooling coil energy - II

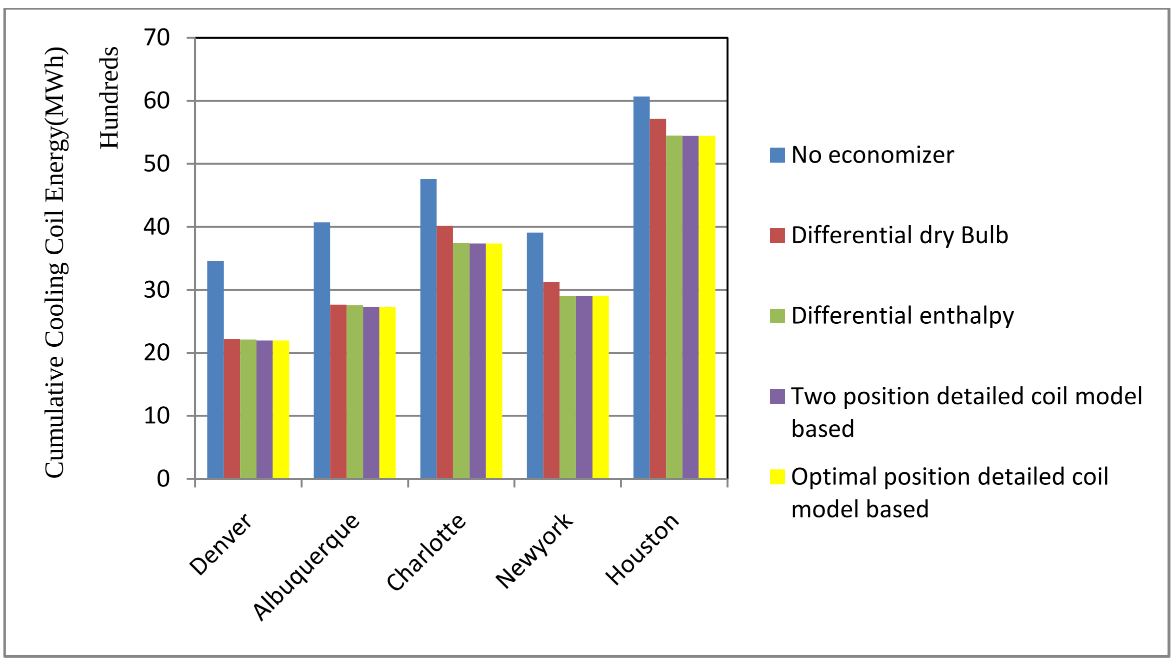


Figure 7.3: Comparison of annual cooling coil energy - III

The results are discussed in the following points:

1. All the economizer strategies showed considerable savings over using no economizer at all under all climatic conditions. However, differential temperature

control predicted higher cooling coil energy in very hot and humid climates than no economizer. This is in accordance with the ASHRAE standards that prohibit the use of differential dry bulb control strategy under those climatic conditions (Table 1.2). This is due to the fact that a differential temperature control would allow 100% outdoor air if it's lower than return air temperature. This type of decision is not based on enthalpy of the air stream. In hot and humid climates like of Miami, for most part of the year, the conditions of outdoor air lie in region C, D and E (Figure 1.2). While the decision in region C and D would be minimum outdoor air which is same as using no economizer at all. However, in region E a differential temperature control would use 100% outdoor air that would end up with more coil load on cooling coil because of more enthalpy. This is why, the cumulative cooling coil energy for climates which are very hot and humid (1A) is more for differential dry bulb than for no economizers.

2. In cities like Phoenix, Cheyenne, Seattle, Sanfrancisco, Denver, Albuquerque and Jackson which lie under humidity classification of either dry or marine, there is less than a 1% difference in annual energy obtained by using differential temperature control over enthalpy based control. The difference in savings range from 0.9MWh to 13MWh.
3. Perfect optimal model based control always gives the least total cooling coil energy in all climatic conditions with perfect sensors that the conventional economizer strategies. The savings, however, are small ranging from 0.75 MWh to 22.4 MWh.

4. Differential enthalpy always performs better than differential dry bulb under humid conditions. The savings in energy range from 129MWh to 273MWh. However, it is not possible to make a clear choice between the two when the climate is dry. Enthalpy based control yields better energy savings in dry climate of Los Angeles than temperature based control.

7.1.2 Hotel and Hospital building

The hotel and hospital buildings differ from office buildings in several respects. The office building does not have as high occupancy as hotel and hospitals which are occupied 24 hours all per day. The HVAC system is continuously in operation in the hotel and hospital buildings to take care of cooling loads at any hour of the day. On the other hand, the office building has a night and weekend set back schedule during which the HVAC system is turned off. As the hotel and hospital building have high occupancy, the minimum outside air required is very high compared to the office building sometimes as high as 100% outside air. Thus, the opportunities for energy savings from the economizer operation is small. The simulation results with perfect sensors for all economizer strategies for hotel and hospital are shown in Figure 7.4 and Figure 7.5 respectively.

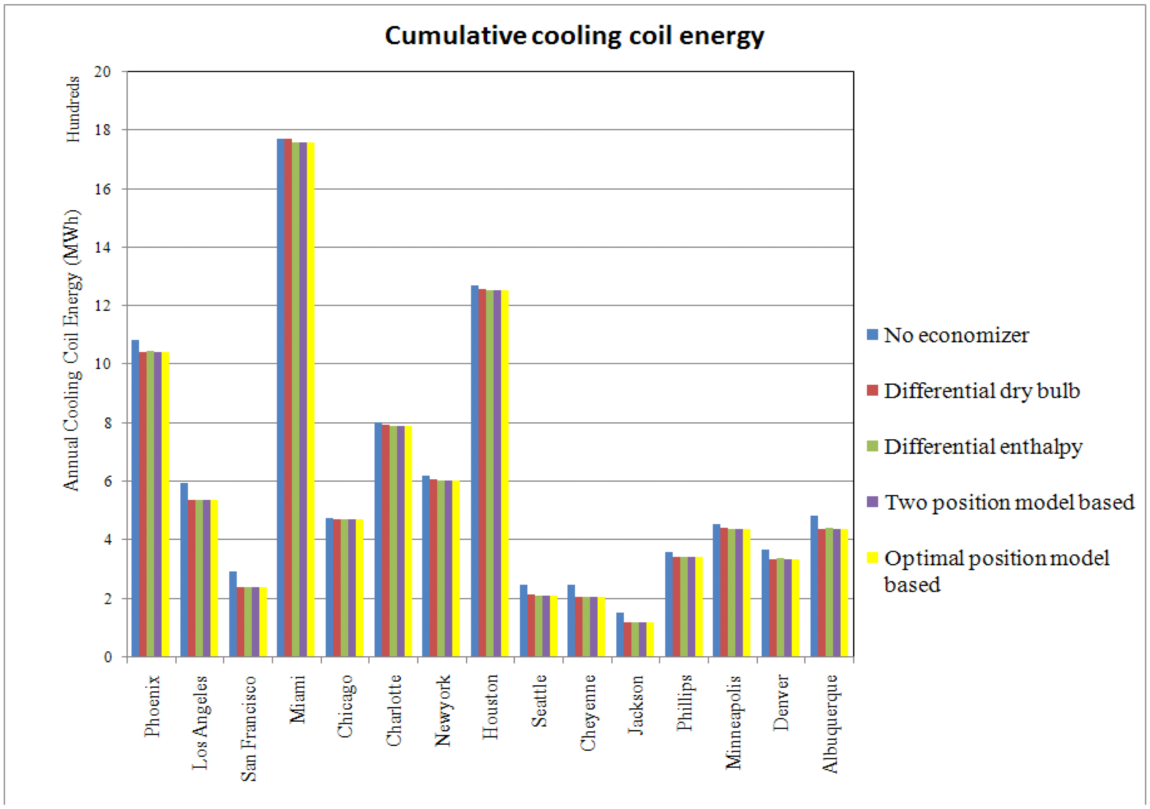


Figure 7.4: Comparison for annual cooling coil energy for Hotel

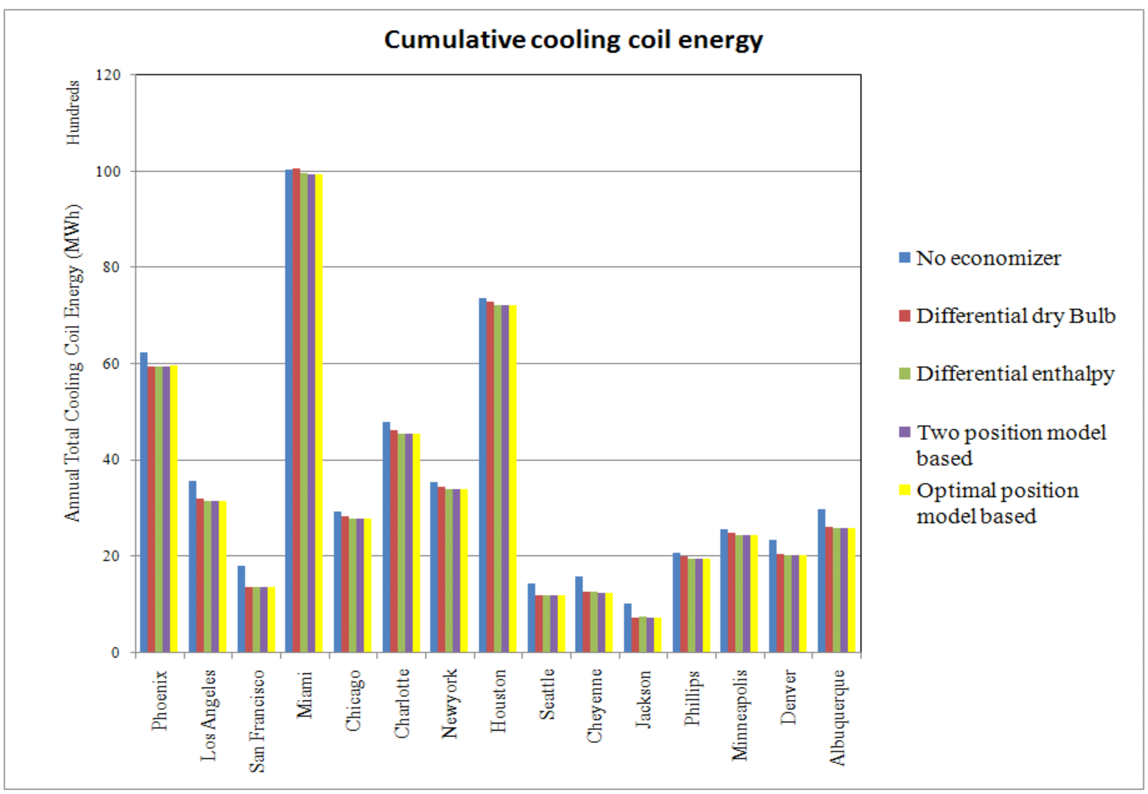


Figure 7.5: Comparison for annual cooling coil energy for Hospital

It can be seen that the percentage energy savings that can be obtained using economizers in these buildings is less than those obtained for office building. The maximum percentage energy savings by using an economizer for a hotel building is 21% and 27% for hospital building in comparison to 60% obtained for an office building. More importantly, these energy savings with economizer strategies could become less or even cease to exist for hotel and hospital buildings when sensor errors are considered. Thus, for buildings which have occupancy schedules like that of hotels and hospitals, it is recommended that a complete analysis of energy savings associated with use of economizer strategies should be done. In the end, it would be just the matter of economics as to whether the controls used for implementing an economizer strategy could justify the cost.

It would be unrealistic to assume that practically economizers can be coupled with AHU's in buildings with perfect sensors. Error in measurements of temperature and relative humidity of outdoor and return air may lead to incorrect control decisions in controller. It is therefore important to see the effect of sensor errors on the control decision. The sensor errors lead to erroneous damper control decision made by the economizer strategy and result in increased coil energy use. In the following sections the effect of sensor error is evaluated for the office building case.

7.2 Imperfect Sensors

The first step was to choose realistic sensor errors on relative humidity and temperature measuring sensors. For this purpose, a sensor manufacturer was chosen that specializes in making sensors for buildings. Airtest, a building sensor manufacturer, has wide range of products including temperature and relative humidity sensors specialized for HVAC equipment. Some of the integrated sensors were reviewed which have both temperature and relative humidity sensor probes and can be placed in ducts. The range of temperature measured is from -40 ^{o}C to 60 ^{o}C with ±0.1 ^{o}C. The relative humidity measuring accuracy is ± 3%. Other sensors found had the range and accuracy close to this data.

To obtain the results with extreme possibilities, reverse sign errors were applied to outdoor and return air. If a positive error was considered for outdoor air then a negative error was considered for return air and vice versa. The same sign error on both the conditions may not produce larger differences.

Sensors have been classified in two categories to see the effects of sensor errors on control decisions made by different control strategies. For the purpose of this study, the sensors with ±1^{0}C error on temperature measurement and ±5% error on relative humidity measurement are termed as well maintained sensors and those with ±2^{0}C on temperature measurement and ±10% on relative humidity measurement are termed as poorly maintained sensors. The large office was simulated for all the five control strategies mentioned above again with four sets of these sensor errors shown in Table 7.1.

The worst scenarios out of the two cases simulated each for well maintained and poorly maintained sensors are picked to derive some general conclusions.

Table 7.1: Sensor errors used in simulations

Case	Temp sensor error	RH sensor error	OA Sign	RA Sign	
I	1^{o}C	5%	_	+	Well Maintained
II	1^{o}C	5%	+	_	
III	2^{o}C	10%	_	+	Poorly Maintained
IV	2^{o}C	10%	+	_	

7.2.1 Results and findings for well maintained sensors

Case I and II from Table 7.1 were simulated for fifteen cities for the large office building. As mentioned earlier, the errors on temperature and relative humidity were only used to make the control decision. After a control decision was made, air loop is simulated with correct values of temperature and humidity. The results for cooling coil energy consumption are shown in Figure 7.6.

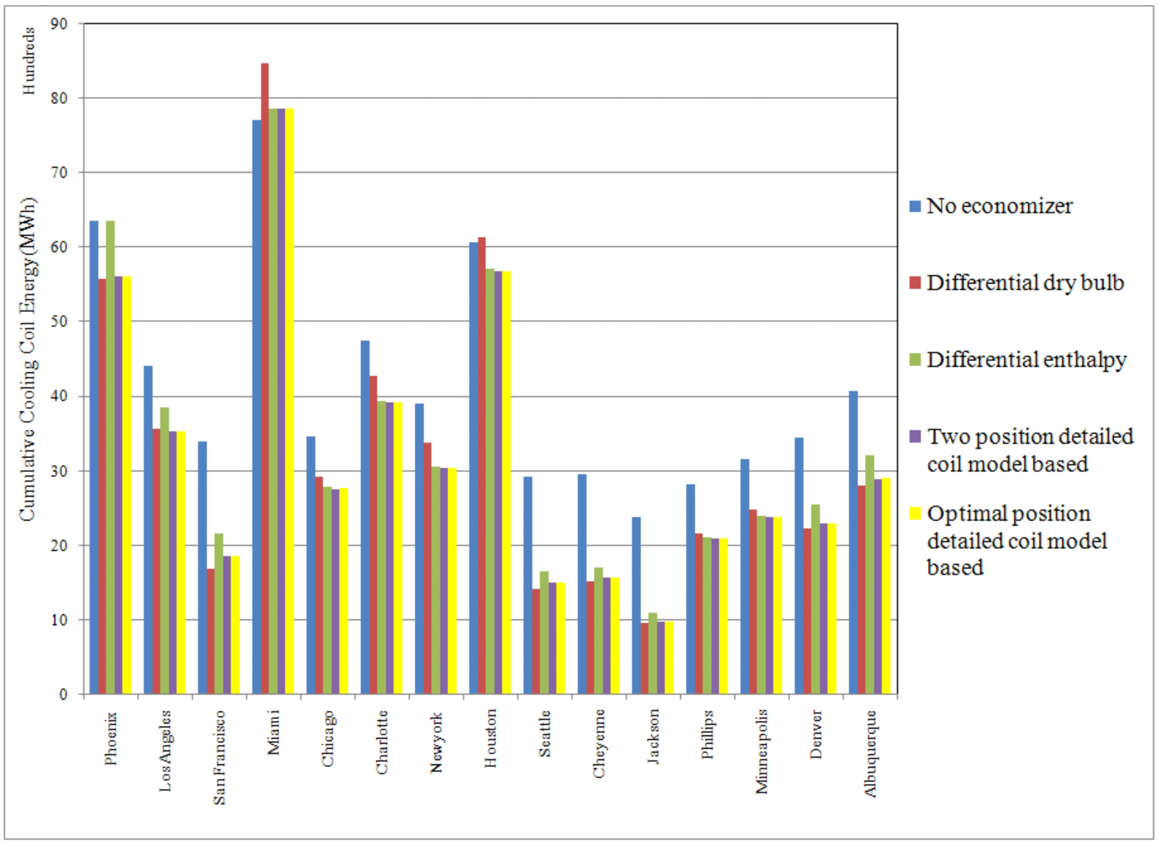


Figure 7.6: Office building, annual cooling coil energy for well maintained sensors

The findings are represented in following points:

1. Significant energy savings are obtained using an economizer even with the sensors errors. The exception is very hot and humid climate of Miami and hot and humid climate of Houston. Temperature based economizer strategy predicts more cooling coil energy for these two conditions.

2. On an average differential dry bulb was the least sensitive to sensor errors of all the economizer strategies. This is due to the fact that all other strategies depend on both temperature and relative humidity sensors. The cooling coil energy lost due to these types of imperfect sensors ranged from 0.4% to 10% of the cooling coil energy used with perfect sensors for temperature based control. On the other hand, it ranged from 1.4% to 16.8% for other economizer strategies.
3. The results for Miami suggest that using any economizer in very hot and humid climates would result in waste of cooling coil energy. However, ASHRAE standard 90.1(2007) doesn't prohibit the use of any other conventional strategy in these climates except differential dry bulb control. The cooling coil energy used when no economizer is used in Miami is 2% lower than the lowest of all the other four economizer strategies simulated.
4. Enthalpy based control continued to perform better in all the humid climate conditions over temperature based control for well maintained sensors. The savings in energy for enthalpy control in humid climates ranged from 48.3MWh to 621MWh over differential dry bulb control. However, optimal position model based control performed only marginally better than differential enthalpy in humid climates which ranged from savings of 12MWh to 32MWh annually.
5. Differential temperature control performed better for all the dry and marine climate conditions. The only exception would be Los Angeles where optimal based control predicts cooling coil energy a little bit better than that predicted by temperature based control. However, the difference is only about 1% (approx) of the energy predicted by optimal based control.

7.2.2 Results and findings for poorly maintained sensors

Case III and Case IV from Table 7.1 are simulated for poorly maintained sensors. The results for annual cooling coil energy are represented in Figure 7.7.

The findings from the results obtained are listed below:

1. Differential dry bulb control still saves cooling coil energy in all climatic conditions except for humid and hot climate of Miami and Houston. The savings associated with differential dry bulb excluding these two climates from observation range from 28MWh to 547MWh.

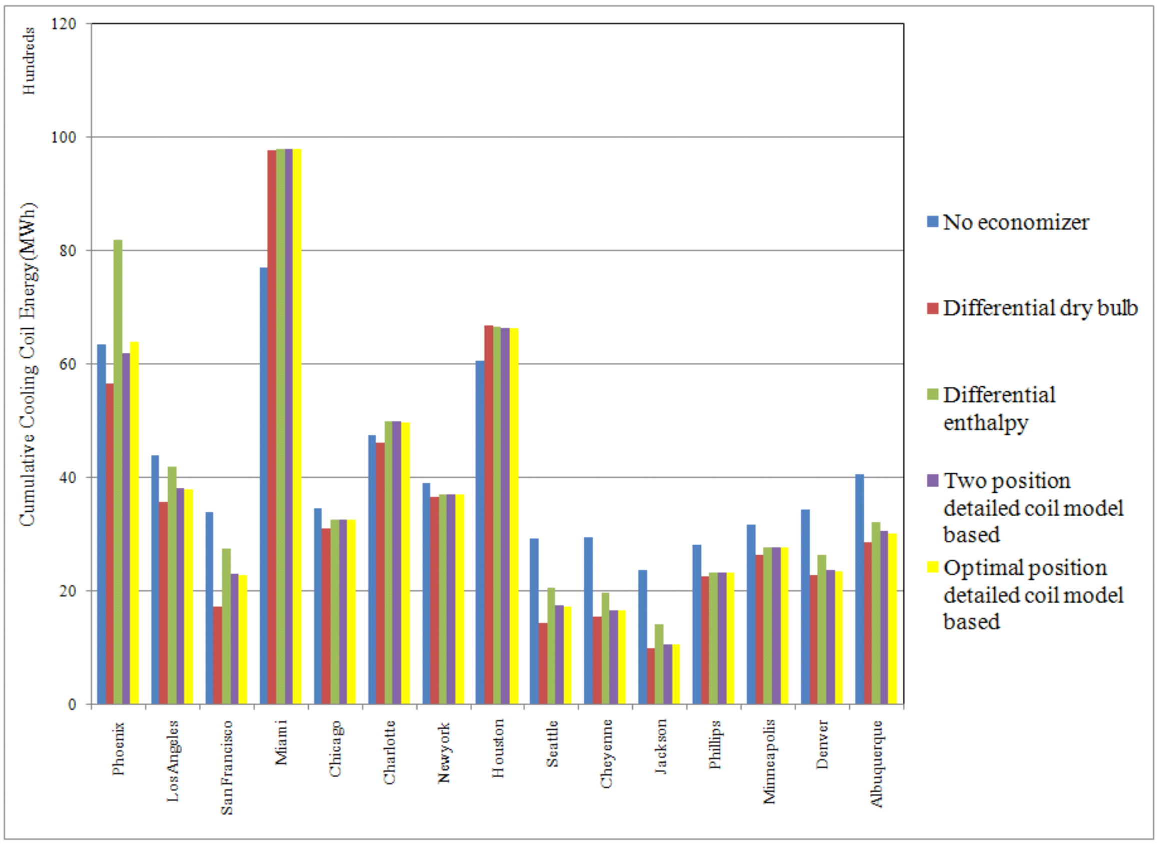


Figure 7.7: Office building, annual cooling coil energy for poorly maintained sensors

2. The cooling coil energy is less sensitive to temperature sensor errors than to relative humidity sensor errors. This again signifies the least insensitivity of differential temperature control to sensor errors. For poorly maintained sensors,

Figure 7.7 shows that no economizers should be used in hot and humid climates to save cooling coil energy.

7.3 Discussion of results

The detailed study of economizer strategies with perfect and imperfect sensors provided a broader understanding of the use of the economizers in real building systems. It was found that modeling sensor errors is an important part of selecting the best control strategies for energy savings. It is thus recommended that building simulation programs should be modified to allow users to specify the sensor errors. ASHRAE 90.1 recommendations for using certain types of economizer strategies in different climatic divisions only hold true with perfect sensors. However, for implementing these strategies in real building management systems, one should consider the sensor errors as well.

Model based economizer strategies could potentially save a lot of energy when used with perfect sensors. Another advantage is that if proper calibration of sensors is done on a timely basis and they are expected to be well maintained, model based an optimal position model based control would work well in every climatic condition providing optimal or near optimal performance. However, for poorly maintained sensors differential dry bulb control could be used in any climatic conditions with exception of hot and humid climates. It is recommended that ASHRAE guidelines for using different economizer strategies in different climatic conditions should be classified on the basis of expected sensor errors.

The study shows that correctly modeling sensor errors is more important than selecting the best economizer strategy. The next chapter discusses the impact of economizer selection on design of the equipment.

CHAPTER 8

8. Impact of Economizer Selection on Design

The standard design practice for sizing the HVAC system cooling coil involves the calculation of the total zone cooling load and adding it to the estimated total outside air load. The coil is then sized to meet this cumulative load. The total outside air load is estimated by using the maximum design day temperature at which minimum fraction of outdoor air would be used for the design calculations. The simulation could be used in two different ways to size the coil. Peak annual cooling coil load can be used for sizing from an annual simulation or a peak design day cooling coil load could be used from a design day simulation. If the outdoor air conditions on the design day are similar to the conditions on the peak annual day, one would expect the two simulation design methods to result in similar coil sizes. However, if conditions differ, coil sizes specified by the two methods are also expected to differ. It is also possible that the peak load hour may occur at an hour where the economizer would call for the outside air to open. This is especially true when the peak load is the morning ‘pick up’ load. The coming sections describe the use of simulation results for sizing and the effect of sensor errors on sizing the cooling coil.

8.1 Overview of simulation based design of cooling coil

The design day conditions used in EnergyPlus are based on ASHRAE design conditions for cooling and heating design day. For example, a 0.4% cooling design day would mean that there are 0.4% of total hours in the year when the temperature of outdoor air is more than the design day maximum temperature. However, the total outdoor air conditions on design day may not be representative of the overall conditions of outdoor air throughout the year. Thus, if a person is using design day simulation results to size the equipment, he might run into risk of over sizing or under sizing the equipment. Consider Figure 8.1 which represents the no economizer cooling coil load on design day for New York on primary y-axis and outside air and return air conditions on secondary y-axis.

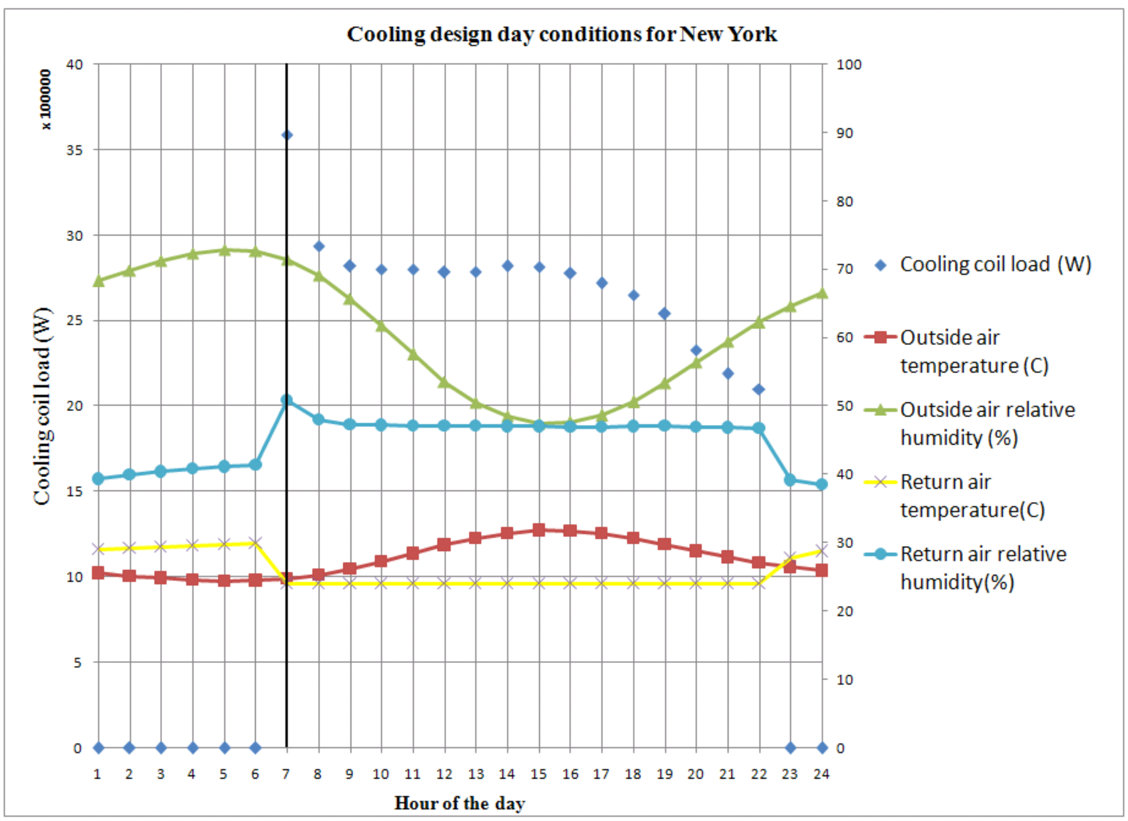


Figure 8.1: Distribution of design day cooling load

As shown in Figure 8.1, the peak cooling coil load occurs early during the day when the HVAC system picks up the start up load. As the HVAC system is off during the night, the return air conditions are allowed to float. The peak load in Figure 8.1 corresponds to the return air and outdoor air with maximum relative humidity for the day. Thus, the mixed air conditions obtained using no economizer at that hour has the highest enthalpy content for the day. Moreover, if differential dry bulb is used than since the return air temperature is slightly greater than outside air temperature, the damper is fully open which again corresponds to maximum enthalpy for mixed air. Both the cases results in highest load on the coil at that hour as shown in Table 8.1. The simulation is performed assuming that the sensors are perfect. It can be seen that if the system is sized according to the peak cooling loads on the design day which does not happen to coincide with the occurrence of maximum temperature of the design day, the system would be over sized.

One key difference between design day calculation and weather file calculation is that the design day calculation is based on ‘steady periodic’ conditions and the annual simulation is not. This does not affect the outdoor air load but it does affect the zone load. In the annual simulation, it is not likely to have multiple extreme weather days in succession. The design day calculation however assumes as many extreme days in succession as required to reach ‘steady periodic’ conditions. This can result in significantly different zone load profiles for peak annual day and the design day.

Table 8.1: Comparison of design day and annual peak cooling coil loads for New York considering perfect sensors

Economizer strategy	DD Peak Hourly Load (kW)	Day	Hour	Annual Peak Hourly Load (kW)	Day	Hour
No Economizer	3585.43	21-Jul	7:00	2412.51	2-Aug	15:00
Differential dry bulb	3691.98	21-Jul	7:00	3038.86	18-Jul	10:00
Differential enthalpy	3585.43	21-Jul	7:00	2412.51	2-Aug	15:00
Two position model based	3585.43	21-Jul	7:00	2412.51	2-Aug	15:00
Optimal position model based	3585.43	21-Jul	7:00	2412.51	2-Aug	15:00

If the design day peak load is used to dictate the sizing of the economizer then considering the annual peak loads for differential dry bulb and other economizer strategies, the equipment might be over sized by 18% when using differential dry bulb control or by 48.6% if any other economizer strategies are used.

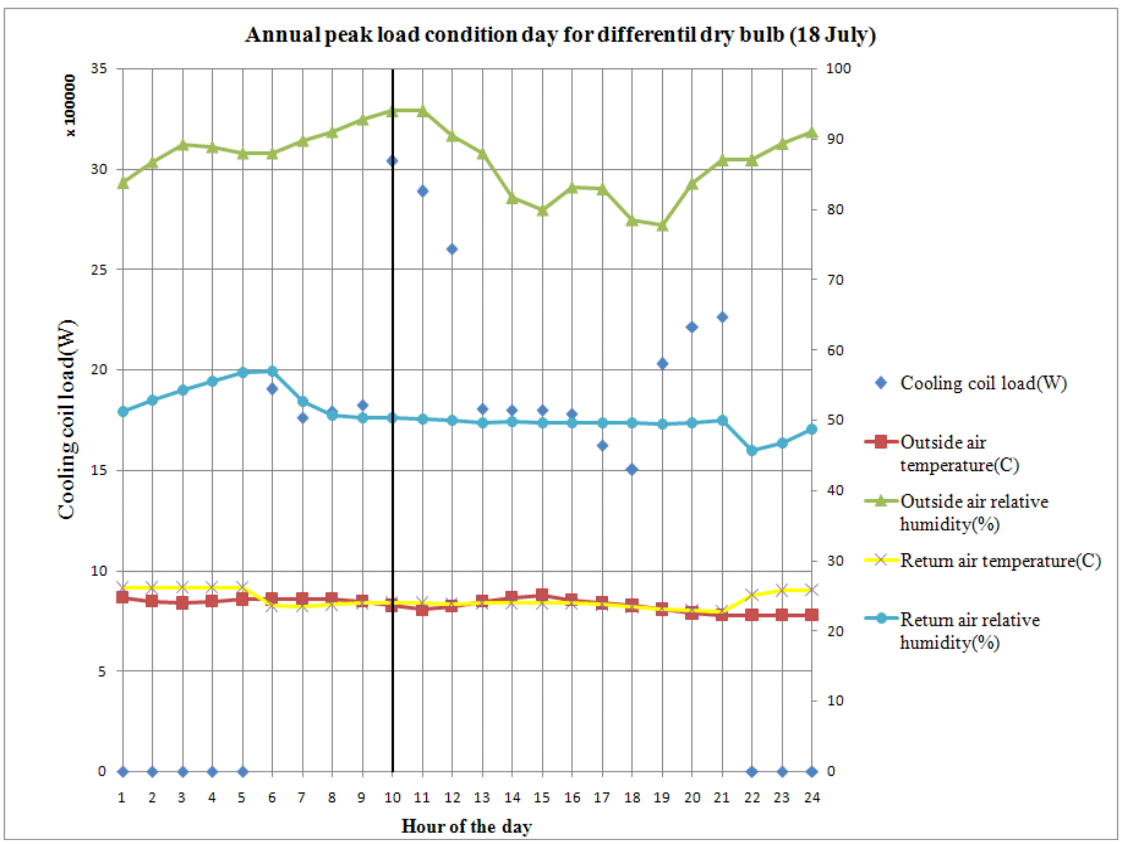


Figure 8.2: Distribution of cooling loads and air conditions on 18 July

As shown in Figure 8.2, the maximum annual peak load for differential dry bulb occurs when the temperature of outside air is slightly lower than return air temperature but the outside air humidity is very high. Differential dry bulb allows 100% outside air in such a case. However, the design day peak load is still greater than the peak cooling coil load obtained during annual simulation with temperature based control. Similar conditions were found for peak load hour for enthalpy and model based strategies as shown in Figure 8.3. Enthalpy and model based economizer closed (minimum position) the damper during peak cooling load hour on 2nd August but the total cooling load on the coil at peak load hour on this day is less than the total load on the coil on design day peak hour.

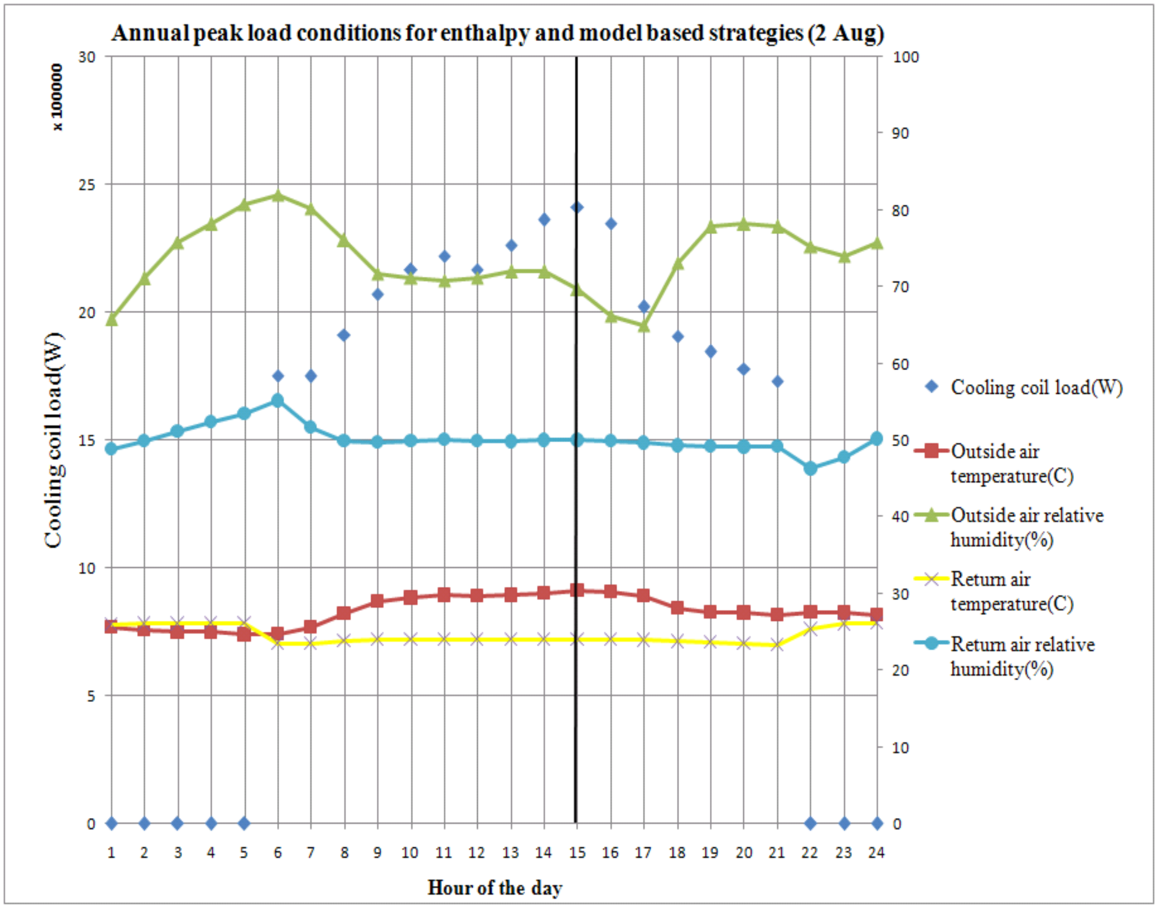


Figure 8.3: Distribution of cooling loads and air conditions on 2 August

Over sizing of equipment would mean that the equipment would mostly run on part-load conditions. Thus, the efficiency of system components would decrease. Also, if cooling coil is over sized, it might not run for full period and thus might not be good for providing zone humidity control. This also increases the initial cost and maintenance cost for the equipment.

Similarly there exists a possibility of under sizing the equipment as well. Figure 8.4 shows the peak design day cooling load for no economizer and cooling loads on the coil for differential dry bulb on annual peak load day.

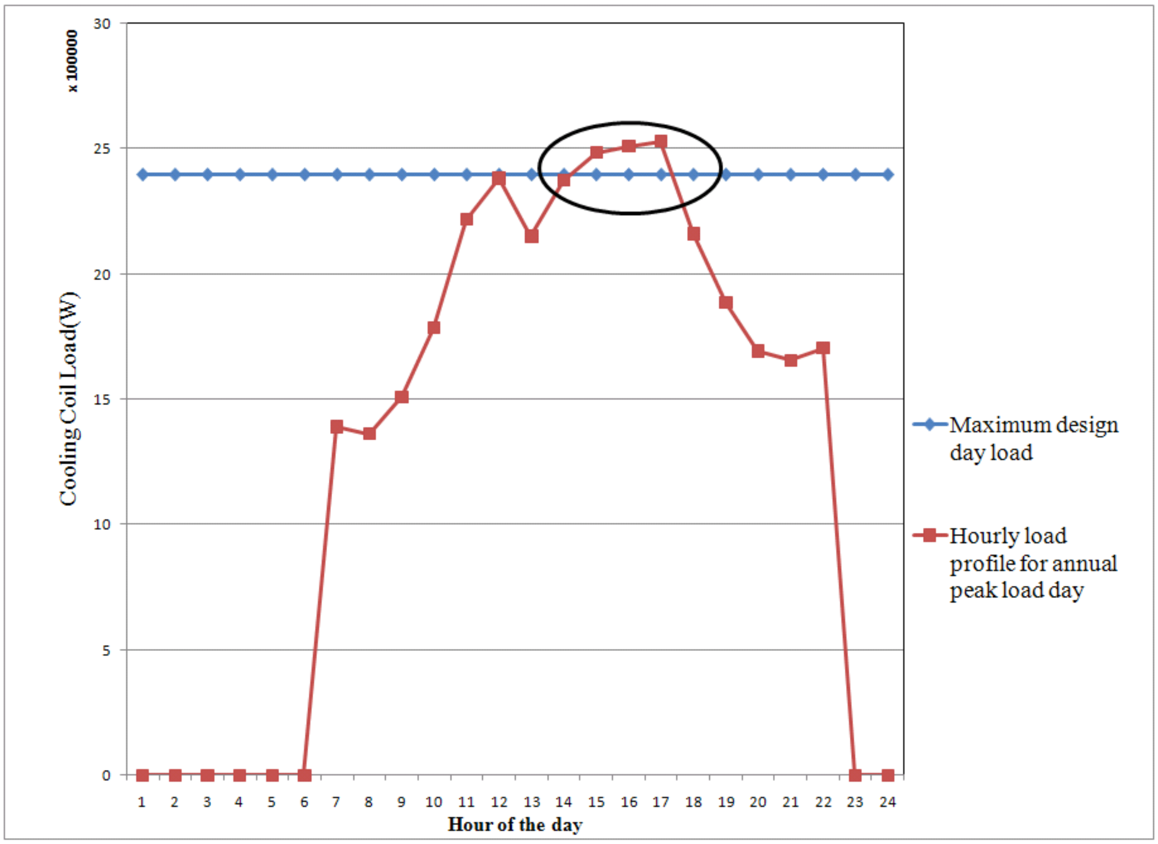


Figure 8.4: Comparison of design day and annual peak cooling coil load

It can be seen that during the three hours highlighted, if the HVAC system was sized based on design day simulation results, the system wouldn't be able to meet the cooling loads. However, the above results are represented considering that sensors are perfect. The number of hours where the system would run into risk of under sized might increase with sensor errors. In an actual building system, the sensors errors might results in bad control decision that could increase the load on the cooling coil. This increase could be less or more than the difference between maximum design day and annual peak cooling coil load obtained assuming perfect sensors. The maximum design day cooling coil load is plotted with the annual peak cooling coil loads for perfect and imperfect sensors for New York. As shown in Figure 8.5 and Figure 8.6, an incorrect control decision would increase the loads on the cooling coil for enthalpy based strategies and differential dry bulb economizer strategies respectively.

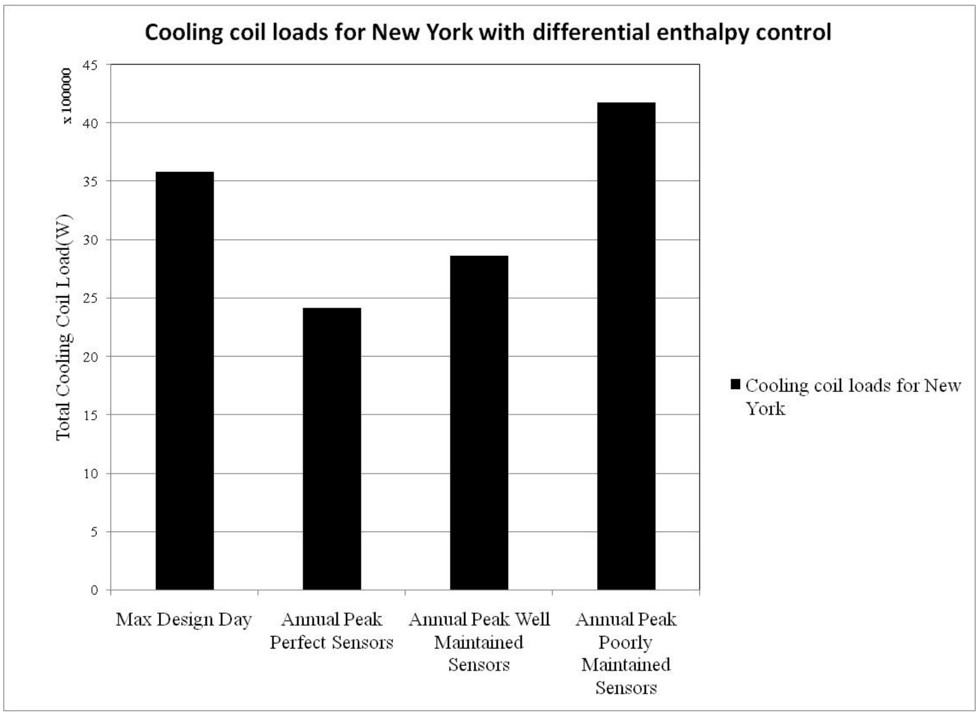


Figure 8.5: Cooling coil loads for New York for differential enthalpy and model based economizer strategies

The cooling coils may or may not be able to meet the cooling load requirements for the increased coil load if sized according to design day maximum cooling load. There are 26 hours for enthalpy based economizer strategies when the cooling load is more than the maximum design day load while there are 9 hours like that for temperature based economizer strategy if the sensors are poorly maintained. A more comprehensive study for all the cities is presented in the next section with peak design day coil loads compared to peak annual loads with and without sensor errors.

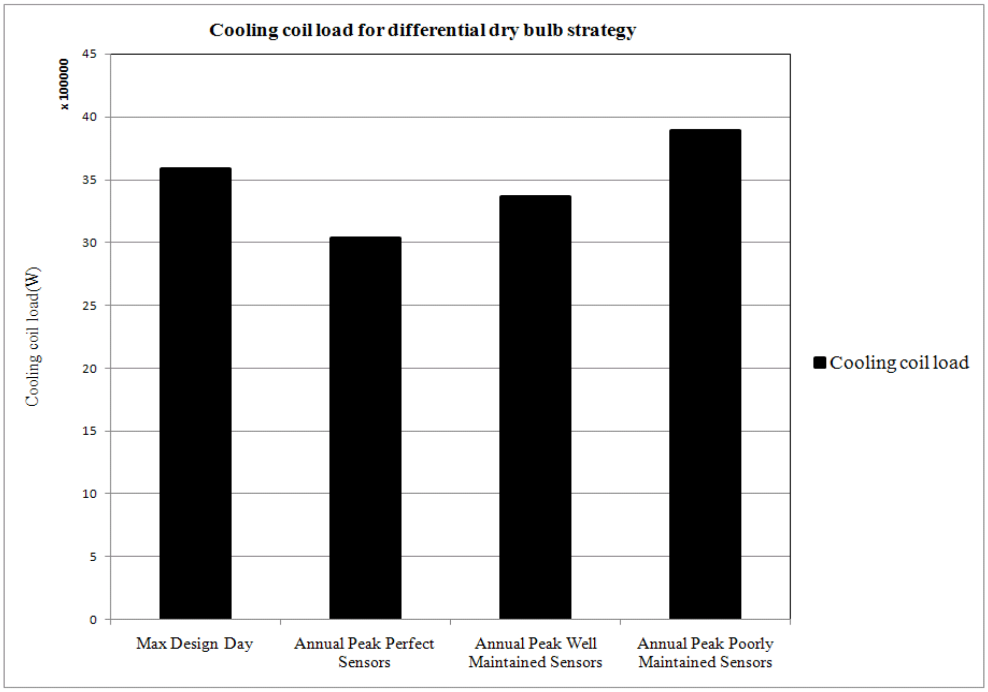


Figure 8.6: Cooling coil loads for New York for differential dry bulb strategy

8.2 Using design day and annual simulation results to size the coil

8.2.1 Perfect Sensors

Figure 8.7 shows the comparison for peak cooling coil loads for design day and annual peak loads for no economizer, conventional economizer strategies and model based economizer strategy. The peak loads for different economizer strategies do not necessarily occur at the same hour of the year. The output obtained in EnergyPlus is an hourly averaged output. There is clearly no single conclusion that can be drawn from the figure.

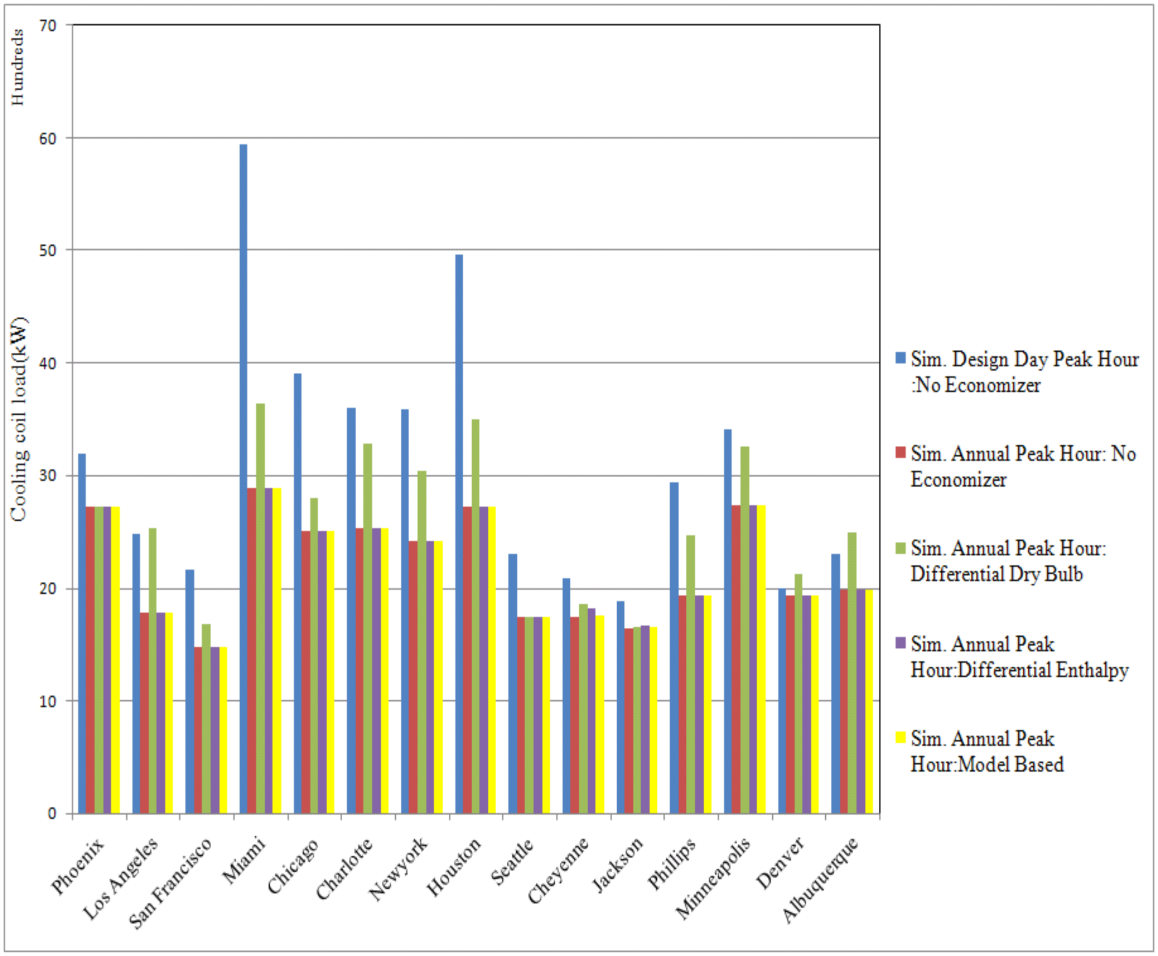


Figure 8.7: Peak cooling coil loads with perfect sensors

The results that can be inferred from Figure 8.7 are:

1. The design day simulation for Miami overestimates the peak annual 'no economizer' and 'enthalpy based economizer' strategies by nearly 100%. That is, if the coil was sized according to the design day simulation and operated with 'perfect sensors' according to TMY weather file, the coil would be double the required size. The other humid climates like Houston show the similar results. The reason for this is that the maximum design day cooling load obtained is the 'pick up' load of the coil. Since the zone is in set-back mode, the return air coming from the zone has high humidity and temperature. Although the

temperature of outdoor air is not the maximum temperature of air during the design day, it still has considerable high humidity. As the system operates on minimum outside air, the mixed air obtained has high enthalpy and humidity content. As a result of this, the latent load on the coil increases and thus the high cooling coil load.

2. Differential dry bulb gives the highest annual peak cooling load in humid climates, ranging up to 41% more than other economizer strategies. On further investigation, it was found that the peak cooling coil loads for differential dry bulb control were obtained when the conditions of outdoor air fell in region E (Figure 1.2). The outside air is cooler than return air in this region but has a high relative humidity. A differential dry bulb control in such a case opens the dampers to allow 100% outside air which is an incorrect control decision. The entering air in cooling coil thus has high enthalpy which considerably increases the load on the cooling coil. Enthalpy controlled and model based economizer strategies close the damper in such a case and thus would not have such higher loads. A similar kind of example is shown in Table 8.2 for New York. The table shows the values of temperature, humidity and loads at the peak cooling coil load hour for differential dry bulb (18th July, 10:00 a.m.) . The values in the column differential enthalpy are picked for the same hour of the year. It is evident that differential dry bulb control makes an incorrect decision by allowing the damper to open fully and increasing the cooling coil load. However, at the same hour differential enthalpy control and model based controls for that matter close the damper and reduce the load on cooling coil.

Table 8.2: Comparison of peak cooling coil loads obtained with differential dry bulb and differential enthalpy at same hour for New York

	Differential dry bulb	Differential enthalpy
Outside air temperature (ºC)	23.5	23.5
Outside air relative humidity (%)	94	94
Outside air enthalpy (KJ/kg)	67.27	67.27
Return air temperature (ºC)	23.8	23.9
Return air relative humidity (%)	50.3	50.2
Return air enthalpy (KJ/kg)	47.16	47.15
Damper position	Open (1)	Closed (0.2)
Cooling coil entering temperature (ºC)	23.5	23.79
Cooling coil exiting temperature (ºC)	10.5	10.5
Load on the cooling coil (kW)	3038.8	1797.4

3. The annual peak loads calculated for differential enthalpy and model based strategies always agrees with, or nearly agrees with the no economizer results. Furthermore, the annual peak hours for these strategies also agree with that for no economizer as depicted for New York in Table 8.1. This indicates that in absence of sensor error, the annual peak hour coincides with the no economizer peak hour condition.
4. If annual simulation results with no economizer are used to size the coil, then there is risk of under sizing the coil for almost every city if differential dry bulb economizer strategy is used.

If the number of hours where an economizer strategy would make incorrect control decision is small compared to the total number of hours of system operation during the year, the error in the annual cooling load calculation will be small (as illustrated in chapter 7). However, if this mistake is made on the sizing calculation it could potentially be a big problem.

8.2.2 Imperfect Sensors

As shown in chapter 7, increasing sensor errors could increase the incorrect control decision which eventually would increase the cooling coil load. Currently sensor errors are not accounted for in either standard sizing procedures or sizing calculations based on simulation codes. A parametric study of comparison of design day peak cooling loads and annual peak cooling loads is done. The results for well maintained sensors are shown in Figure 8.8 and those for poorly maintained sensors are shown in Figure 8.9.

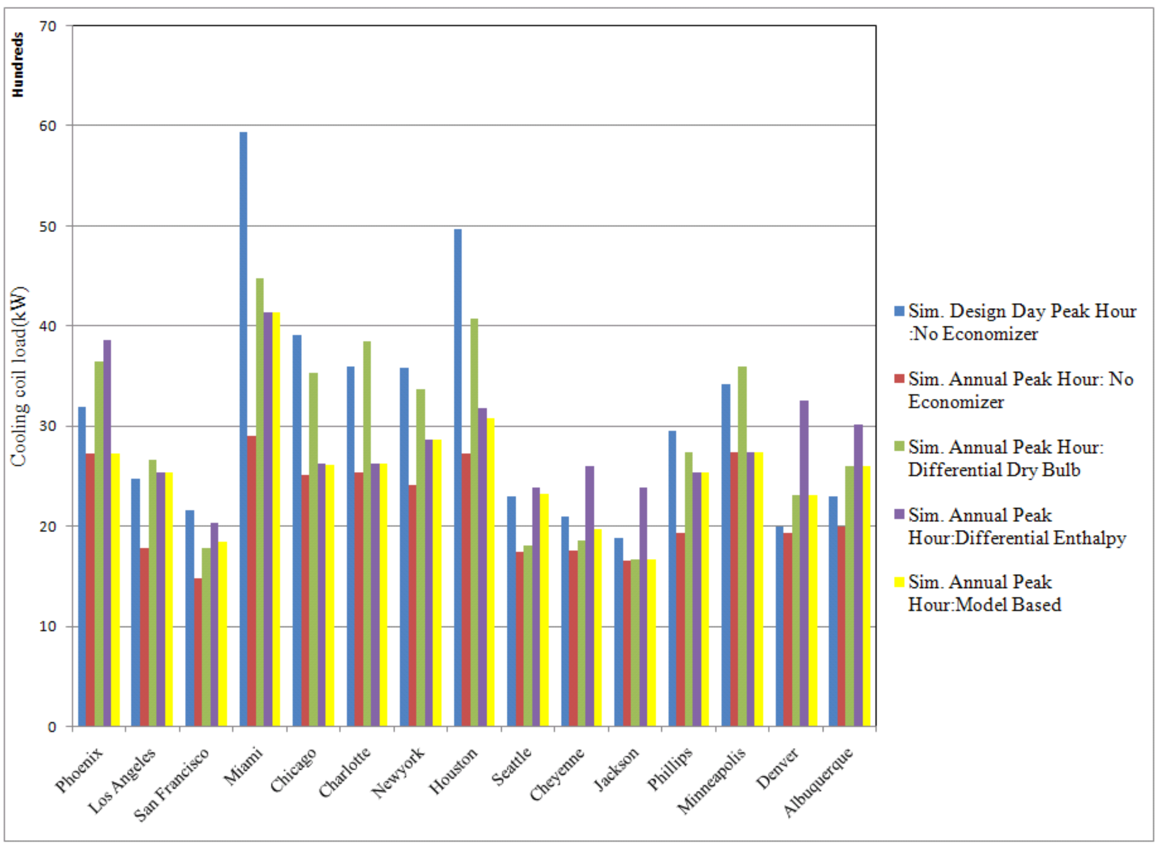


Figure 8.8: Peak cooling coil loads with well maintained sensors

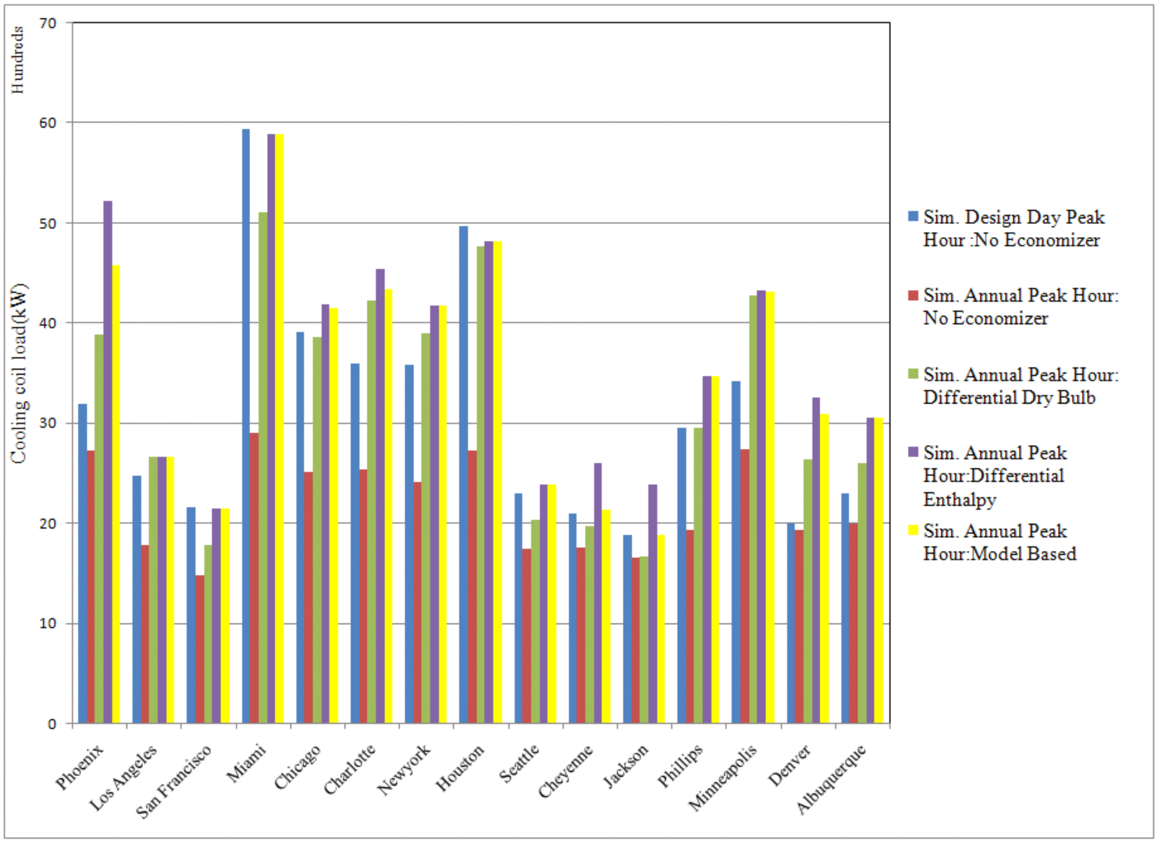


Figure 8.9: Peak cooling coil loads with poorly maintained sensors

A designer using annual simulation results to size the equipment would expect that a conservative design would dictate sizing with no economizer. This is because he would normally expect the peak hour to occur at extreme outdoor air conditions. For these conditions the outside air damper should be at minimum position which is the case with no economizer. It was found with simulations done with perfect sensors that using the design day peak load for sizing could result in over sizing of the coil. However, if the sensors are well maintained, the annual peak loads obtained come closer to the expected peak loads. If the coil is sized based on design day peak loads, there exists a risk of under sizing the coil even with well maintained sensors for any economizer strategy used in cities like Denver, Albuquerque and Los Angeles as shown in Figure 8.8. Two out of three economizer strategies would mean the same thing for Phoenix and Seattle.

Moreover, if the sensors are imperfect, sizing the coil based on annual peak load obtained for no economizer would certainly undersize the coil at all places for any economizer strategy. Furthermore, if the sensors are poorly maintained, then the cooling coil would certainly be undersized for any economizer strategy as shown in Figure 8.9. In such a case turning the economizer off is the best option.

A designer with a plan to implement any economizer controls for the AHU should consider the effect of sensor errors in sizing calculation. As shown in Figure 8.8 and Figure 8.9, the conservative way of sizing may not be helpful at all. The cooling coil system sized like that could run into risk of not meeting the cooling requirements during some hours of the year.

8.3 Recommendations

A designer should carefully investigate the weather file for a location if using design day maximum load for sizing the equipment as the design day conditions may not be representative of conditions for the year. The designer should use the simulations tools to identify the annual peak cooling loads in addition to maximum design day cooling coil loads. Simulation tools should be modified to account for user defined sensor errors. A designer could use such a tool to determine the effect of sensor error on the cooling coil loads in a year and can use the data to design the equipment for efficient use.

Considering that the best performance achievable for the economizer strategies would be with well maintained sensors, it is recommended that design day peak load could be used for sizing for all the humid climates. The exceptions being Charlotte and Minneapolis where differential dry bulb strategy would have unmet cooling load hours if implemented. However, the coils for Houston and Miami would be still over sized by almost 40% using design day peak load for well maintained sensors. It is recommended that design day peak loads should not be used for sizing for dry and marine climatic conditions. Moreover, if the sensors are poorly maintained, design day peak loads should never be used for sizing the coil. The exceptions in this case again are Miami and Houston which would have almost a perfect coil size for all economizer strategies.

CHAPTER 9

9. Conclusions and Recommendations

The purpose of this research was to identify a better control strategy for economizer that could work fine irrespective of any climatic conditions and give more savings in terms of cooling coil energy use. A modeling strategy that uses minimization of total cooling coil load was developed for damper control at outside air and return air inlet. Intensive simulations were performed by developing the code in EnergyPlus. The simulations were performed over three benchmark building type file obtained from DOE website. Evaluation and verification of the conventional strategies and new modeling strategies was done with results obtained from Seem and House (2009). Impact of sensor errors on decision making ability of the economizer strategies was also studied. The parametric study was performed over these three files to determine cooling coil energy use and annual peak cooling coil loads. Impact of economizer selection on the cooling coil energy and design of the equipment is also discussed. The important conclusions based on the results obtained and recommendations for using these economizer strategies for saving cooling energy and designing the system coil are listed in coming sections.

9.1 Conclusions

Important conclusions obtained through the work done are listed below:

1. In office buildings, model based strategies could give optimal or near optimal performance and save cooling energy given if the sensors are well maintained. The only exception is very hot and humid climate of Miami where using no

economizer is the best option. The savings associated with model based strategy for perfect and well maintained sensors range from 5% to 60%.

2. If the sensors are poorly maintained then differential dry bulb economizer strategy would provide the most energy savings. The exception to this rule again is again the hot and humid climates of Miami and Houston.
3. Incorrectly estimating the bypass factor in the controller model does not introduce significant error in the annual cooling coil energy estimation as long as the bypass factor is reasonable. The percentage change in cooling energy obtained with change the bypass factor from 0.05 to 0.25 was less than 1%.
4. In buildings like that of a hotel or hospital, which have high occupancy and require more outside air for ventilation purposes, using any economizer strategy does not make a large difference in saving any cooling energy .
5. Hoping that the best available sensors that would be used in building management systems would be well maintained sensors, sizing the coil based on design day peak load would always undersize the coil for dry and marine climatic conditions for one or more economizer strategies. Moreover, in cases like that of Miami and Houston, there is risk of over sizing the coil by 100% for perfect sensors and about 40% for well maintained sensors. A conservative approach of sizing the coil using the peak annual load for no economizer would always undersize the coil regardless of the economizer strategy used.

9.2 Recommendations

Several recommendations can be made for a designer using the simulation code for evaluating the use of economizer strategy and for the actual implementation of the economizer controls in real building systems. The major recommendations are listed below:

1. It is recommended to use no economizer in very hot and humid climates as of Miami as even with well maintained sensors, other economizer strategies use more energy than no economizer strategy.
2. Proper maintenance and calibration should be done for sensors installed in building systems to realize the energy savings that could be obtained with economizer strategies.
3. ASHRAE standards should be modified to recommend economizer strategies for different climates based on quality of sensors used.
4. It is recommended that a designer should carefully analyze the annual and design day peak cooling loads to design the equipment. Furthermore, proper consideration should be given to effect of sensor errors on sizing the equipment.
5. For total analysis of loads and energy, simulation codes should be modified to take into account the effect of sensor errors in a real building management system.
6. The EnergyPlus simulation assigns the entire 'pick-up' load to the first time step of operation. If the time step is short, the 'pick up' load could be extremely high. The simulation should be modified to allow the user to select the time period over which the 'pick up' load should be applied.

CHAPTER 10

10. References

Avery, G. (1984). "VAV Economizer Cycle: Don't use a Return Fan." Heating, Piping, and Air Conditioning 56(8): 91-94.

Brandemuehl, Michael J. and Braun James E. (1999). "The Impact of Demand Controlled and Economizer Ventilation Strategies on Energy Use in Buildings." ASHRAE Transactions 105(2).

Budaiwi, I. M. (2001). "Energy Performance of the Economizer Cycle under Three Climatic Conditions in Saudi Arabia." International Journal of ambient energy 22(2): 83-94.

Carrier,W.H. (1937). "The Contact-Mixture Analogy Applied to Heat Transfer with Mixtures of Air and Water Vapor." Transactions of The American Society of Mechanical Engineers.

Crawley Drury, P. C., Linda Lawrie, Winkelmann Frederick, (2000). "EnergyPlus: Energy Simulation Program." ASHRAE Journal (April): 49-56.

Dickson, D. K. (1986). "Economizer Control Systems." ASHRAE 28(9): 32-36.

Elmahdy, A. H. and Mitalas, G.P. (1977). "A Simple Model for Cooling and Dehumidifying Coils for Use in Calculating Energy requirements for Buildings." ASHRAE Transactions 83(2): 103-117.

Hittle, D. C., and Johnson D.L. (1985). "New Control Design Principles Based on Measured Performance and Energy Analysis of HVAC Systems." Interim Report E-85/02, USA-CERL.

Kao, J. Y., and Pierce, E.T. (1983). "Sensor Errors:their Effects on Building Energy Consumption." ASHARE Journal Dec. 1983: 42-45.

Kays, W. M., London, A.L. (1964). "Compact Heat Exchangers." 2nd Edition McGraw-Hill:New York.

Khutoryanskiy, Leonid and Margadant,G. S.(1999). "Optimizing an Airside Economizer." Heating, Piping and Air Conditioning: 35-38.

Seem, J.E., House, J. M. (2009) "Development and Evaluation of Optimization-based Air Economizer Strategies." Applied energy (accepted for publication).

Spitler, J. D., D.C.Hittle, D.L. Johnson and C.O.Pedersen (1987). "A Comparative Study of the Performance of Temperature-Based and Enthalpy-Based Economy Cycles,." ASHRAE Transactions 93(2): 13-22.

Wacker, P.C. (1989). "Economizer Savings study." ASHRAE Transactions 96(1): 47-51.

Yiu, J. C. M. (2000). "Assessment of Practical Applications of Outdoor Air Economizer in Hong Kong." Building services engineering research & technology 21(3): 187-198.

Zmeureanu, R. (1989). "A Case study of Energy Savings due to the Economizer System in Montreal." Energy 14(9): 537-544.

VITA

AMIT BHANSALI

Candidate for the Degree of

Master of Science

Thesis: DEVELOPMENT AND ANALYSIS OF MODEL BASED ECONOMIZER STRATEGIES

Major Field: Mechanical Engineering

Biographical:

Personal Data: Born in Bhopal, Madhya Pradesh, India on December 17, 1984

Education:

Received the B.E. degree from Institute of Engineering and Technology, Indore, India, 2007, Mechanical Engineering
Completed the requirements for the Master of Science with a major in Mechanical Engineering at Oklahoma State University, Stillwater, Oklahoma in July, 2009.

Experience:

Worked as an engineering intern at IIT Kanpur, India. Involved in making the experimental setup to measure effect of roughness on pressure drop across mini channel. Worked as a research assistant during graduate school. Performed code checks for EnergyPlus and maintained the documentation. Added some new modules in EnergyPlus. Served as Teaching assistant for first year in graduate school. Helped students to learn EES and Solidworks.

Professional Memberships:

ASHRAE national and student member
Vice President of ASHRAE student branch June 2008 - Present

Name: Amit Bhansali Date of Degree: December, 2009

Institution: Oklahoma State University Location: Stillwater, Oklahoma

Title of Study: DEVELOPMENT AND ANALYSIS OF MODEL BASED ECONOMIZER STRATEGIES

Pages in Study: 97 Candidate for the Degree of Master of Science

Major Field: Mechanical Engineering

Two new model based economizer strategies were developed and implemented in the EnergyPlus simulation program. Model performance was verified by comparing EnergyPlus results with previously developed models. Results agreed within ±5%. Both detailed and bypass coil models were evaluated for potential use in the model based economizer strategies. Energy and peak load analyses were performed for three benchmark buildings using five economizer strategies for 15 US climatic conditions. The impact of sensor errors on energy savings and coil design was also discussed. For office type buildings, it was found that model based economizer strategies do have energy savings potential as long as sensors are well maintained. However, if sensors are poorly maintained, differential dry bulb was found to be the best strategy. For Hospitals and Hotels with high minimum outside air requirements due to high occupancy, the economizer strategies resulted in lower energy savings. The bypass coil model was found to be in agreement with results obtained for detailed coil model based strategies. Using different simulation methods to size the cooling coil can change the coil size by up to 100%. Accounting for sensor error in simulation based coil design, tends to decrease the difference between the design day and annual peak cooling coil loads. Based on the results of the study, it was recommended that simulation capabilities should be modified to account for user specified sensor errors and ASHRAE standards should be modified to characterize economizer use based on building use and qualities of sensors.

ADVISER’S APPROVAL:____